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WELDED METAL BRIDGES.

Comparison between the German and the French standard specifications.

(Case of the plate girder bridges)

by H. REGIS,

Engineer, « Office Technique pour l'utilisation de l'acier (O.T.U.A.), France. »

(Annales des Ponts et Chaussées, France.)

INTRODUCTION.

The technique of assembling metal bridges by electric arc welding has developed to an extraordinary degree in Germany in the course of the last few years. Although welding was taken up later in Germany than in the United States, which was the pioneer country in this particular field, Germany has now undoubtedly assumed the leading place in so far as the welding of bridges is concerned.

In France, on the contrary, the development of welded bridges is comparatively slow, in spite of the fact that the French builders, not less than the engineers of the Administrative Services, are anxious to hasten this development. In particular France has not yet set forth in systematic form the advantages, as compared with riveting, which welding would appear to present according to the claims made by Germany.

Amongst the reasons which have been put forward to explain this divergence of opinion as between the two countries, the view has been expressed that the French specification was particularly strict; it is evidently an argument which, owing to the very nature of their profession, builders of metal structures will readily endorse, and the Technical Office for the Utilisation of steel (O. T. U. A.) has thought it of interest, therefore, to ascertain exactly how far this is true.

It is an indisputable fact that the French and German specifications have been drawn up on the basis of very different data and information.

In Germany, where the specification for welded metal bridges has already been revised on four occasions, the present official requirements are derived from a methodical and exhaustive series of experimental tests carried out over a number of years and bearing almost exclusively on the resistance which the industrial ferrous metals offer to the fatigue resulting from alternating stresses, a type of stresses having no doubt little effect on road bridges, but which must be taken into account when considering railway bridges.

Since, in France, exhaustive information based on experiment was not available, the engineers responsible for drawing up the French standard specification have not had any important laboratory information at their disposal for the preparation of this document. It is not to be wondered at, therefore, that these

engineers, when deciding the exact details of the specification (for which the heavy responsibility rests with them) should protect themselves by prudently imposing strict requirements.

This then is the question which the O. T. U. A. has decided to investigate impartially, and, let it be said, in no disparaging spirit, by means of a detailed comparison of the two above-mentioned specifications.

A difficulty arose at the very outset. The two specifications, which seem to speak different languages, are derived from basic principles which on the surface at any rate would appear to be different; it seems essential therefore to proceed by adapting language and symbols in such a manner that a clear and simple comparison may be made. With the help of certain hypotheses, and by the elimination of certain terms of little importance, it has appeared possible to transpose the French specification in a general way by adopting the form of the German standard specification. How this has been done is explained in the following pages.

The considerations which follow are restricted to plate girder bridges, as these are the only type dealt with by the present German specification.

Similarly the so-called high-resistance constructional steels (St. 52 and St. 54) have been disregarded, the inquiry being limited to the ordinary steels (St. 37 and St. 42).

I. — *Materials.*

We have set out below in tabular form the different characteristics of the French and German specifications in so far as the choice and control of the parent and deposited materials are concerned.

	FRENCH SPECIFICATION (St. 42 and St. 54).	GERMAN SPECIFICATION. (St. 37 and St. 52.)
Tests on parent metal and deposited metal.	The tests comprise: Tensile tests on test pieces entirely composed of deposited metal; Tensile tests on test pieces with welded joint. Bending tests. No cross weld test. (Cross weld tests are used when examining welders.)	The tests comprise: Tensile tests on test pieces entirely composed of deposited metal. Tensile tests on test pieces with welded joint. Bending tests. Cross weld test. (No distinction between tests for welders and tests of materials.)
Minimum tensile strength . .	Respectively 42 and 54 kgr./mm² (26.7 and 34.3 tons/sq. in.).	Respectively 37 and 52 kgr./mm² (22.9 and 33 tons/sq. in.).
Maximum tensile strength . .	Respectively 50 and 64 kgr./mm² (31.75 and 40.6 tons/sq. in.).	(No maximum tensile strength.)
Minimum elongation .	Respectively 18 to 20 %.	(No minimum elongation.)
Minimum resilience .	8 kgrm./cm² (373 ft.-lb./sq. in.).	(No minimum resilience.)
Endurance	(No endurance limit.)	Respectively: 15 and 16 kgr./mm² (9.5 and 10.2 tons/sq. in.) for transverse welds (not machined), 18 and 19 kgr./mm² (11.4 and 12.1 tons/sq. in.) for transverse welds (machined), and for longitudinal welds (after 1.8×10^6 pulsations).

The German specification does not enforce, as does the French, tests of the deposited metal each and every time that a new work is undertaken. A particular type of electrode is tested once for all by an official laboratory at intervals of two years. The constructor therefore has only to satisfy himself that the electrodes he uses conform exactly with the one which has passed the official test.

II. — *Fatigue of welds.*

In both specifications the maximum permissible fatigue for the deposited metal is obtained by multiplying the permissible stress for the parent metal by a coefficient α which varies according to the nature of the weld and the type of stress applied.

NATURE of the weld.	TYPE of stress.	COEFFICIENT α. French specification. In the workshop.	French specification. At the site.	German specification. In the workshop.	German specification. At the site.
Butt weld.	Tension	0.85	0.75	0.75	Not laid down.
	Compression	0.95	0.85	0,85	Not laid down.
	Bending	0.95	0.85	0.80	Not laid down.
	Shear	0.65	0.55	0.65	Same value as in the workshop.
Other welds.	All types of stress.	0.65	0.55	0.65	

III. — *Permissible stresses in the case of variable loads.*

The determination of the permissible stresses when the loads are variable differs greatly in the two specifications. The German specification takes into account factors which are ignored in the French, and which we will first of all summarize (1).

1. *German definitions.*

The following considerations, definitions, and notation form the basis of the German standard specification.

Let us consider a test piece of metal submitted to loads of variable magnitude on a pulsating machine. Let σ_u represent the absolute value of the minimum limit of the unit stress (kgr./mm²) resulting from these loads, and let σ_o represent the absolute value of the upper limit of this unit stress. Further we can distinguish the one value of these unit stresses from the other by an algebraic sign, namely plus (+) in the case of tension and minus (—) in the case of compression.

When σ_u and σ_o are of similar sign, we speak of *increasing* loads; when they are of opposite sign, we speak of *alternating* loads.

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(1) Summary based on KOMMERELL's : « Notes concerning the requirements for welded structures in the case of railway plate girder bridges » (Berlin, 1936), the German notation being retained.

The interval $\sigma_o - \sigma_u$ is called the *range* of the alternation or of the pulsation.

WÖHLER and his successors have demonstrated that for every value σ_u of the lower limit of the pulsation, and for each value $\sigma_o - \sigma_u$ of the *amplitude* of that pulsation, there exists a number « N » of repetitions (range of fluctuation) above which number the test-piece invariably breaks. This number of repetitions before rupture « N » increases when the amplitude, i. e. the upper limit σ_o diminishes. When this number « N » attains the value 2×10^6, in many cases the test piece is unlikely to fail, even were the number of repetitions to be indefinitely increased.

The value of σ_o corresponding to this number of repetitions before rupture 2×10^6, is called the endurance limit, corresponding to the lower limit of unit stress.

In all that follows the symbol σ_o will systematically represent the endurance limit thus defined.

The endurance limit σ_v or σ'_v corresponding to a lower limit of unit stress of zero value ($\sigma_u = O$) bears the distinctive name of *primitive strength* or « repetition limit » stress (1).

The endurance limit σ_w such that it shall be equal in value but opposite in sign to the corresponding lower limit ($\sigma_o = -\sigma_u$) is called the *reversal limit* (1).

The results thus arrived at can be easily represented graphically (WEYRAUCH), by plotting the values of the minimum limits σ_u as abscissæ and the corresponding values of the endurance limits as ordinates, taking into account as usual the signs given to either of these two quantities.

By drawing the two bisectors of the co-ordinate axes it is seen that for every value of σ_u, represented by the abscissa OI (fig. 1), there is a corresponding length JK set off as ordinate; this length JK represents the permissible range or amplitude for the corresponding oscillation.

Should the endurance limits σ_o be tensions, a graph such as that shown in Fig. 1 is obtained, in which the extent of oscillation is marked by vertical hatching.

(1) Cf. MORLEY's « Strength of Materials ».

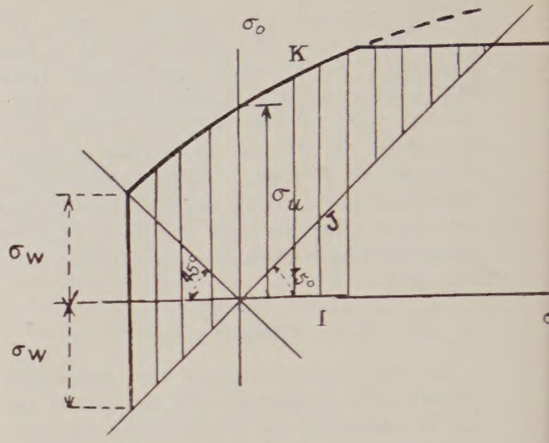

Fig. 1.

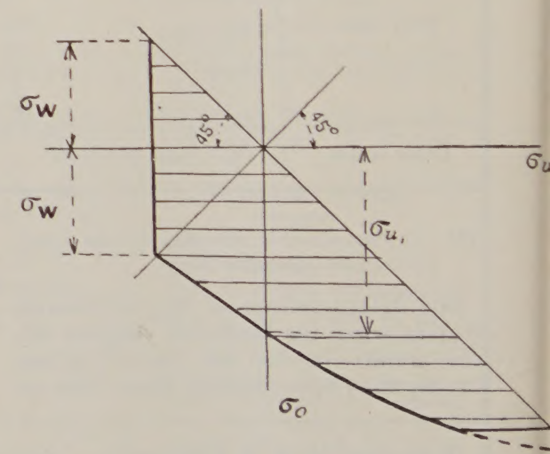

Fig. 2.

In the case of the endurance limits σ_o in compression (fig. 2), the zone of oscillation is shown in horizontal hatching.

In practice the curves of the σ_o—'s are broken on the right side and do not exceed in height a line parallel to the axis of the abscissæ and distant from this axis by a quantity σ_F, which corresponds to the case where the values of the endurance limit reach the zone of permanent elongation for tension, or permanent crushing for compression.

On the left side, the curves of the σ_o—'s are limited as abscissæ to the value of σ_w, equal to the alternating strength.

For all practical purposes it can be assumed that the two portions of the curves (either for tension and for compression) bounded on the right by the horizontal limit σ_F, and on the left by the vertical limit σ_v, may each be replaced by two straight lines, which are not a production of one another, but diverge to the right and left by the values σ_v and σ'_v of the reversal limit in tension and in compression (fig. 3).

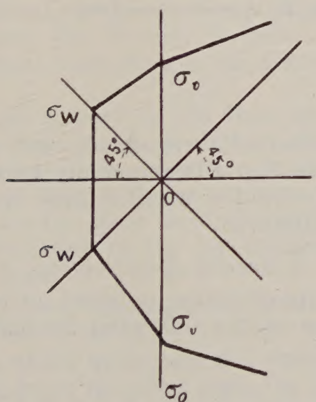

Fig. 3.

2. *German standard specification.*

This specification covering welded railway bridges is defined graphically by a certain number of diagrams, separately plotted for each of the different varieties of commercial constructional steel.

Each of these diagrams is plotted in accordance with the definitions and the method of experimentally measuring the terms σ_u and σ_o which have just been described.

The numerical values taken for the term σ_o, from which the graphs in question are derived, are not the exact values of σ_o obtained by experimental measurements, but the experimental values systematically reduced by 1 kilogramme per square millimetre ($\sigma_v = \sigma_o - 1$).

The factor of safety given thereby results primarily from this reduction of 1 kilogramme/mm² (0.63 ton sq./in.), and is further enhanced by the fact that the experimental diagrams have been plotted for 2×10^6 alternations, whereas in actual practice the bridges will not be called upon to resist more than a quarter of this number of alternations in their lifetime.

There is naturally a diagram for the parent metal, and several other diagrams each corresponding to a definite type of weld.

Fig. 4 represents the various diagrams referring to the current German commercial steel (called St. 37); each of these diagrams corresponds to the following material and types of weld — the symbol *a* corresponds to the values σ_D in tension, and the symbol *b* to the values σ_D in compression :

Diagram I. — Parent metal. (i. e. unjointed plate).

Diagram II. — Butt-welded joint with a back run applied to the root of the weld, and the welds machined.

Diagram III. — Butt-welded joint, without a back run being applied.

Diagram IV. — Joints with front fillet welds, or near the extremities of side fillet welds, the edges of which have not been machined.

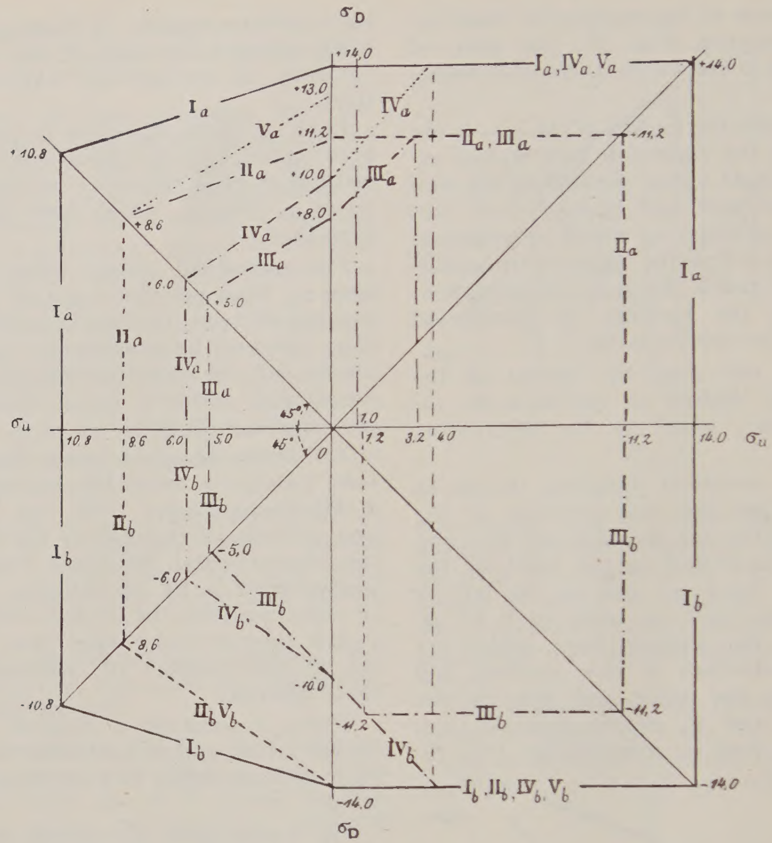

Fig. 4.

Diagram V. — Joints with front fillet welds, or near the extremities of side fillet welds, the edges of which have been carefully machined.

It should be noted in this graph that careful grinding of the edges of welds produced a marked increase in resistance. A comparison of Graphs II and III for butt welds, and Graphs IV and V for fillet welds, makes this immediately apparent.

Note furthermore, the superiority of very good fillet welds over very good butt welds (V and II) in the case where the absolute value of the maximum stress is tensile.

By virtue of their shape, each of the graphs in Fig. 4 is commonly known in Germany by the name « little house » (das Häuschen).

3. *French specification.*

This specification is based on the observation of the following inequalities :

(1) $c' + d'_1 + 0.5\,(d'_1 + d'_2) + t' \leqslant \alpha R_1$,
(2) $c' + d'_1 + 0.5\,(d'_1 + d'_2) + v' \leqslant \alpha R_2$,
(3) $c' + t' + w' \leqslant R_2$,
in which :

c' = Stress induced by the dead load;

d'_1 = Maximum stress, of similar sign to c', produced by the live load;

d'_2 = Absolute value of the maximum stress, of opposite sign to c' (if there is no stress of opposite sign to c', d'_2 will be taken as equal to zero);

v' = Stress induced by a wind of 150 kgr./m² (30.7 lb./sq. ft.) force;

w' = Stress induced by a wind of 250 kgr./m² (51.2 lb./sq. ft.) force;

t' = Stress due to the action of the temperature = Coefficient of reduction due to the weld; ;

R_1 and R_2 = Permissible limits of stress, depending on the nature and quality of the metal used.

In the case of French steel, so called « Ponts et Chaussées » quality (i. e. Ac. 42) which is the quality under review, the specification requires :

Tension and compression :
R_1 = 13 kgr./mm² (8.25 tons/sq. in.);
R_2 = 14 kgr./mm² (8.9 tons/sq. in.).

4. *Transposition.*

In order to make a simple and comprehensible comparison of the permissible stresses given by the French formulæ with those authorized by the German standard specification, let us try to represent the French formulæ in graphic form according to the German method.

Let us remark in the first place that the periods of the oscillations due to the temperature are infinitely longer than those of the oscillations due to the live load. On the other hand, the wind effect is considered in the calculations as a static load.

It is not unreasonable therefore to consider the terms v', w', and t' as being of the same nature as the stress c' resulting from the dead load, for the purpose of the comparison in question.

We must, however, be careful when making this comparison to consider in the French specification a 42 kgr./mm² steel as against a 37 kgr./mm² steel in the German specification. The rates of the permissible stresses R_1 and R_2 should therefore be proportionately reduced : R_1 from 13 to 11.45 kgr./mm² (8.25 to 7.27 tons/sq. in.) and R_2 from 14 to 12.33 kgr./mm² (8.9 to 7.8 tons/sq. in.).

These proportionate reductions are essential if one is to arrive at a fair comparison of the requirements of the two specifications.

Having said this, let us note that the above quoted inequality [1], concerning tension may be written :

$$(c' + t' + d'_1) + 0.05 \times [(c' + t' + d'_1) - (c' + t' - d'_2)] \leqslant \alpha R_1,$$

If we remark moreover that the French expressions

$$c' + t' + d'_1$$

and

$$c' + t' - d'_2$$

are none other than the German expressions designated by σ_D and σ_u we may conclude that the above mentioned inequality may be written :

$$\sigma_D \geqslant \frac{2\alpha R_1 + \sigma_u}{3} \qquad R_1 = 11.45$$

which transcribes in German terms the requirements of the French standard specification.

Applying similar treatment to inequality (2), we are able to write :

$$\sigma_D \geqslant \frac{2\alpha R_2 + \sigma_u}{3} \qquad R_2 = 12.33.$$

If we reproduce these two inequalities in graphic form by means of a Weyrauch diagram we obtain a series of « little houses » whose outlines vary according to the values of α.

In Fig. 5 have been shown only the graphs corresponding to the more fa-

[1] Strictly speaking one should take into account the variations of t'. But in general they are sufficiently slight to be ignored, in comparison with the variations in stress due to the live load.

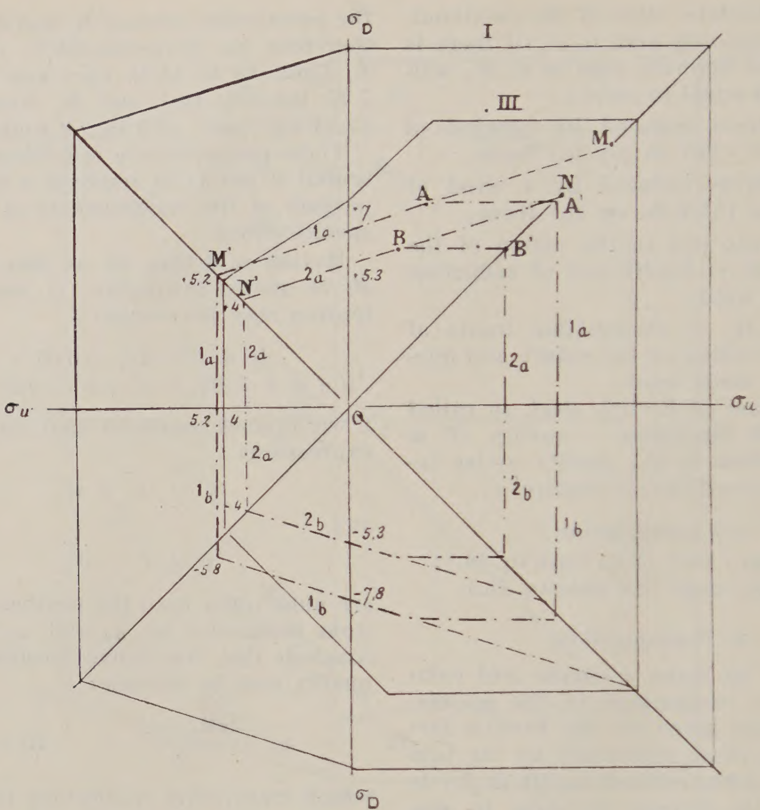

Fig. 5.

Key : Full lines represent graphs derived from the German specifications, corresponding to the parent metal (Graph I), and the poorest type of weld (Graph III).

vourable inequality, i. e. the second, with $R_2 = 12.33$.

These graphs are drawn to exactly the same scale as those of Fig. 4.

Diagram I corresponds to butt-welds.

Diagram II corresponds to fillet welds.

This latter diagram is dissymmetrical with reference to the axis of the x—es, as is to be expected, since α does not keep the same value when changing from tension to compression.

It should be noted that the above-mentioned equations only give the oblique portions (MM' and NN') of the « little houses ». The horizontal portions AA' and BB' are determined by the fact that d'_2 is considered as zero if there is no stress of sign opposite to c'. This means that commencing with a certain value of σ_u equal to $c' + v'$, the permissible σ_D keeps a constant value equal to :

$$\sigma_D = \frac{2\alpha R_2 + c' + v'}{3}.$$

5. *Comparison of the two specifications.*

It can be concluded from a casual glance at the comparative graphs in Figs. 4 and 5, that the French specific-

ation acts as a serious deterrent to builders of welded bridges.

Welding is burdened not only with the general coefficient of reduction α, but further with the term 0.5 $(d_1 - d_2)$ which occurs in the formulæ, and which moreover is not found in the general formula of the French standard specification for riveted bridges (1).

In addition the French standard specification does not take into account the numerous results of endurance tests which prove that a grinding of the welds after deposit substantially increases the resistance to fatigue of the welded joints by doing away with the « notch » effect particularly injurious in stresses of this character.

Whatever the cause the best of welds in France is considered nearly inferior to the worst of welds in Germany.

It is worthy of note that the graphs derived from the French specification present approximately the same shape (the scales being similar) as the experimental diagrams of the German specification. This circumstance, the result no doubt of a fortunate coincidence, may serve as a help towards bringing into line the requirements of the two specifications.

Thanks to the obligations imposed in France as regards quality alike for parent metal, electrodes, and the standard of workmanship of the welders, it is unlikely that the welds made in our workshops are inferior to those made in the German shops (always excepting the careful grinding of the welds after deposit). Under these circumstances there would appear to be ample grounds for relaxing the strictness of the French standard specification.

With this end in view, whilst adhering to the method of notation by arithmetical inequalities, to which the French engineers are accustomed, one could :

Either replace the term 0.5 $(d'_1 - d'_2)$ by another function of these two quantities, which would be of lower value, and which should furthermore keep the transposition diagrams similar in shape to the German diagrams; in this case the values of the coefficient α would remain unchanged. This is an investigation which we have not undertaken;

Or, preserve in their existing form the first members of the inequalities (1) and (2), whilst replacing the coefficient α in the second member by a new coefficient β, the numerical values of which would be suitably adapted to the various cases.

Considering this second method of transposition and taking a value $\beta = 1$ in the case of butt welds, the immediate result is that the transposition diagram derived from the French formulæ can very nearly be superimposed on the German diagram corresponding to butt welds of the poorest quality.

Modifications of this nature may possibly be criticised on the ground that they are purely empirical. However, this objection should have little weight provided such modifications lead us to results which are being constantly verified in actual practice. In fact, the German standard specification which has proved satisfactory in spite of its liberality, can scarcely claim to have been arrived at by any other method.

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(1) For this latter type of bridge, the formulæ of the French standard specification are:

$$c + d + t \geqslant R_1.$$
$$c + d + t + v \geqslant R_2.$$
$$c + t + w \geqslant R_2.$$

In these inequalities the symbols have the same meaning as the same symbols (with accent) occuring in the specification for welded bridges.

[625. 113]

Railway Curves. — Junctions.

Problems of layout. — Junctions with transitioned turnout-curves

by J. CHAPPELLET,

Assistant Engineer, Central Permanent Way Service, French National Railways Company (Northern Area.)

(1) High speeds over the through and branch roads.

Let us consider the case of a junction in which the through roads AB, CD, of the two turnouts are laid on the straight, and examine a vehicle which travels over the two branching-off roads AE, CF.

This vehicle, when on the left-hand track, passes first over the straight portion MA, then the branch AE, in the facing direction, then the connecting curve EG, and then the diamond-crossing GH, (which we will assume to have different angles).

On the right-hand track the vehicle traverses the branch road CF, in this track, in the trailing direction, and afterwards the straight portion CP.

It is obvious that the passage of this vehicle at high speed will be comfortable only if the path which it pursues contains no objectionable changes of curvature, so that the resulting changes of acceleration are imperceptible to passengers standing in the corridor of the vehicle.

In order to arrive at such an alignment, there are three conditions which must be satisfied.

It is necessary in the first place for the radii of curvature of the turnout AE, of the connecting curve EG, and of the diamond-crossing, either to be equal, or, for their differences to be such that :

$$f_2 - f_1 \leqslant \frac{8.6}{v^2}$$

f_2 and f_1 being the versines of the two adjacent curves, and v the speed in metres per second [1].

Further, the shock of entry into the curve, at A and at F, and of exit at H, and at C, must not give rise to jolts detrimental to the comfort of passengers. Finally, the switch angle of each turnout should be zero.

We will now consider how far it is possible to satisfy these various requirements.

In order to reduce the shock of entry into the curve at A, there are two possible solutions; either, make the radius of the turnout curve equal to at least 2 000 m. (100 chains), or let the turnout curve become a parabolic transition, the curvature of which is zero at A and equal to $\frac{1}{R_1}$ at E ; R_1 being the radius of the curve EG.

The first solution has the drawback of necessitating an appreciable change of radius at E, as the radius of the curve EG cannot as a rule be made more than 500 to 700 m. (25 to 35 chains) on account of the rather restricted track intervals (six-foot way) usually encountered.

[1] See *Bulletin of the Railway Congress*, October 1930.

If the radius of the curve through the diamond crossing is in the neighbourhood of 1 000 m. (50 chains) and the radius of the turnout curve is to be in the neighbourhood of 2 000 m. (100 chains), the curve EG must take the form of a parabolic transition, the curvature of which will vary from 1/2 000 to 1/1 000, and the average curvature will be 1/1 330, which will still require too great a track interval.

We are of the opinion, therefore, that the second solution is to be preferred and that the alignment of the turnout curve of the connection between the left-hand roads should be a parabolic transition.

In order to eliminate the shock of exit from the curve at H, it is sufficient either to let the diamond crossing be followed by a curve of the same radius as that of the diamond crossing itself, or to suitably connect the curve of the diamond crossing either to the straight or to some other curve of appreciably different radius which may follow it.

Adopting for the connection in the right-hand roads a turnout curve of radius at least equal to 2 000 m., the shock of entry at F, when the turnout is preceded by a length of straight, and the shock of exit at C, will both be negligible. In the case where the turnout curve is either preceded or followed by a curve of radius appreciably different from 2 000 m., the turnout and this curve should be joined by a transition.

So far as the switch angles are concerned the construction of the switches does not admit of these being made zero. They should, however, be reduced as far as possible. The values selected depend partly on decisions regarding the clearances required between stock rail and switch tongue (the length of the latter being restricted to about 14 or 15 m. (46 to 49 ft.), and partly on the thrust produced when a vehicle traverses them at high speed, the magnitude of this thrust being measured by means of accelerometers.

These considerations led us, at the end of 1930, to entertain the idea of a junction with a turnout road taking the form of a parabolic transition (fig. 15), the calculations for which are given at the end of this article (Note 1).

This simple idea, logical though it is, now that there is no longer any question of the necessity for parabolic transitions in main line curves was not pursued at an earlier stage owing to the belief that the switch angle [1] alone was responsible for the objectionable jolt at the entry to the switches.

Now, it is also possible to conceive that the shock of entry into a curve, of average radius [2] without parabolic transition, may in fact be the principal cause of this jolt.

We would remark that, so far as we are aware, measurements of accelerations at switch angles and at the tangent points of un-transitioned curves, as part of the study of switches and crossings, had not previously been attempted at that time. And yet accelerometers had been available for a long time, notably the Auclair (mass-type) which would have enabled such information to be obtained with quite sufficient accuracy. As an experiment, a transitioned turnout with a switch angle of 0° 45' 0" was designed, (although the standard switch angle of 0° 51' 34" was retained for general use). Firstly it was ascertained experimentally

(1) The switch angle in use on the System at that time, for junction switches, was 0° 51' 34".

(2) The radius in use on the System at that time, for non-symmetrical turnouts, was 736 m. (37.8 chains).

that a switch tongue 12 m. (39 ft. 4 1/2 in.) long would enable the requisite flangeway clearances between tongue and stock rail to be maintained, with a single stretcher bar for driving purposes.

With the object of improving the riding at speed over the junction at Longueau (Signal Box No. 1), the two connections of the double junction, having turnout curves of 736 m. (36.8 chains) radius, a switch angle of 0° 51' 34", a crossing angle of tan 0.06, arranged as shown in Fig. 1, were replaced by two connections with transitioned turnouts, in the same position.

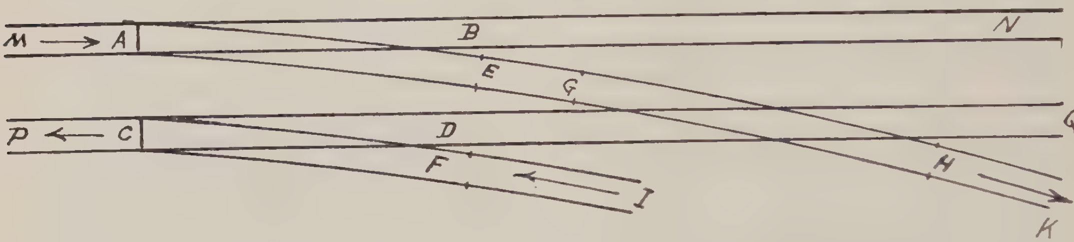

Fig. 1.

In order to make a comparison between these turnouts, tests, involving measurements of acceleration, were made at high speeds on the junctions with circular turnout curves and on those with parabolic transitions. We will now set forth, and analyse, the results of these tests.

Running tests at high speeds on circular turnout curves of 736-m. (36.8 chains) radius (1) of the tan 0.06 connections of Longueau Junction (Signal Box No. 1).

In the first place we would draw attention to the fact that the switches of the turnouts with circular curves being superelevated 0.09 m. (3 1/2 in.) throughout their length were preceded, on the straight portion, by a cant developing gradient 125 m. (410 ft.) long, on which the superelevation increased at the rate of 1 in 1 000, being eased off slightly at the toe of the switches.

Running on the left-hand road. — Speed 100 km. (62 miles) p. h. — Switch taken facing.

The rearmost vehicle, in which the Mauzin-Langevin piezo-electric quartz accelerometer was placed, was a coach of wooden construction, with bogies 13.68 m. (44 ft. 10 in.) apart, and weighing 40 tons, and the accelerometer was stationed over the rear bogie. On account of the cant gradient on the straight track in front of the switches, on which the 90 mm. (3 1/2 in.) of superelevation is attained, the vehicle is in contact with the running edge of the low rail, from the point where the superelevation reaches 70 mm. (2 3/4 in.) up to the toe of the switch tongue (Fig. 2) (1). Between the moment the vehicle runs over the toe of the tongue, and the moment when the shock of entry into the curve makes itself felt in the body of the coach, a cer-

(1) In practice 736 m. (36.8 chains) is only a theoretical figure, owing to imperceptible deformations; the radius of the turnout curve may be appreciably less than this at certain points.

(1) The acceleration curves have been plotted to the following scales : 1 mm. to the metre (1 in 1 000) for distances g = 23 mm.

tain interval must elapse while the play of the wheels on the track and the lateral play between the bogie and the coach are taken up.

About 4 m. (13 ft. 1 1/2 in.) from the toe of the tongue, at the end of the short length of straight at the toe and tangent to the curve of 736 m. radius, the coach body is subjected to an initial acceleration of approximately $g/4$ which rises to $g/3 = 2.50$ m. (8.2 ft.) beyond. *Then, in 1/30 sec., the acceleration changes abruptly from g/4 to g, representing a change in acceleration of 22g per second.*

This is the effect of the shock of entry into a curve without parabolic transition. Damping takes place rapidly. Thereafter, on account of the lack of superelevation for the speed and radius of the curve, the vehicle clings to the outer rail, due to the centrifugal force which fluctuates about a value of $g/2$.

Running on the right-hand road. — Speed 110 km. (68.3 miles) p. h. — Switch taken trailing.

The rearmost vehicle, in which the Mauzin-Langevin piezo-electric quartz accelerometer was placed, was a coach of wooden construction, with bogies 13.68 m. (44 ft. 10 in.) apart, and weighing 40 tons, and the accelerometer was stationed over the rear bogie. After traversing the switch (Fig. 3) the wheels which are running on the high rail, strike the latter at a point 8 m. (26 ft. 3 in.) beyond the toe of the switch by reason of the change of direction, by which point the lateral play between the high rail and the components of the vehicle has been taken up.

This thrust gives rise to an acceleration of $g/2$. The vehicle then rebounds on to the low rail which it strikes in a

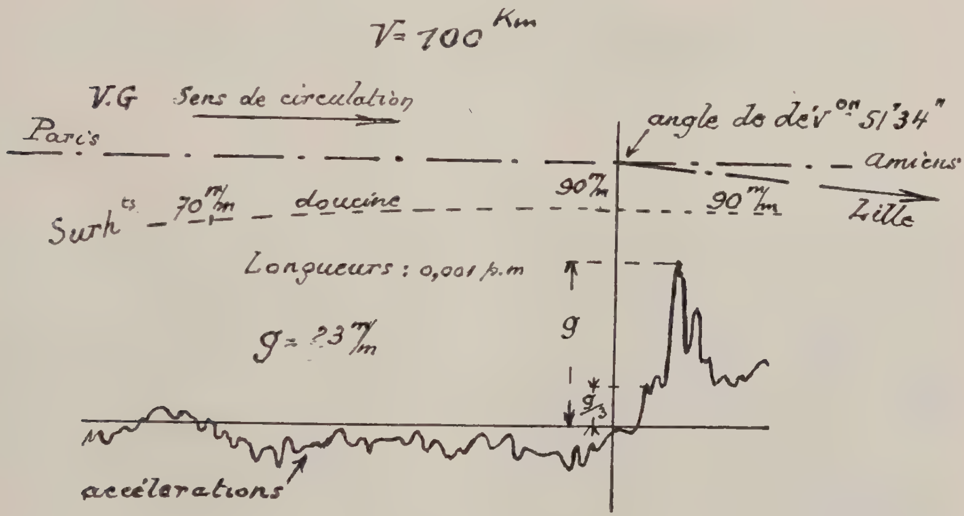

Fig. 2.

Explanation of French terms:

Sens de circulation = direction of travel. — Angle de dev^{on} = switch-angle. — Doucine = easement. — Surh^{ts} = superelevation. — Longueurs : 0,001 p. m. = distances, 1 in 1 000 scale.

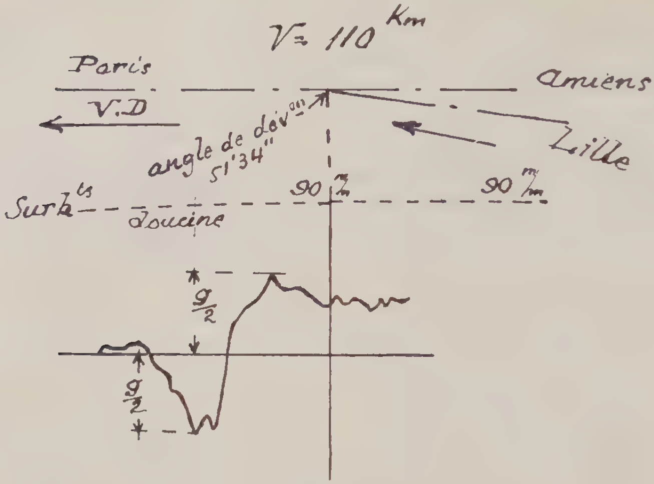

Fig. 3.

similar manner with an acceleration of $g/2$ approximately.

This hunting motion dies away rapidly. In addition, at the exit from the turnout curve, the sudden removal of the centrifugal force and of the superelevation of 90 mm. (3 1/2 in.) on the straight portion, sets up in the suspended coach body a rolling motion, as seen in the extract from the Hallade record (Fig. 4) which reveals an oscillation of 60 mm. (2 3/8 in.) amplitude.

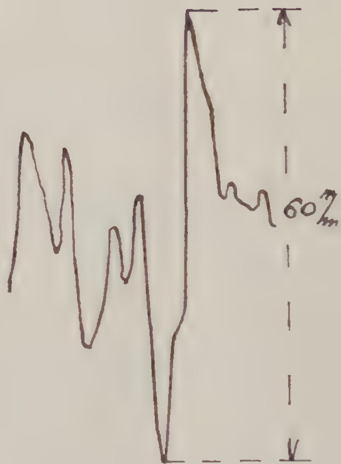

Fig. 4.

Running tests at high speed over the parabolic transitions of the new turnouts at Longueau Junction (Signal Box No. 1).

We will first describe the characteristics of the turnout with parabolic transition. This turnout of a total length of 62.68 m. (205 ft. 8 in.), consists of a switch with tongues 12 m. (39 ft. 4 1/2 in.) long, the switch angle being equal to 0° 45' 00". The turnout curve conforms to a transition varying from radius ∞ at the origin to 810 m. (40.5 chains) at the end. The crossing, of manganese steel, is of the « Est », double-angle pattern, angle tan 0.045 on the wing-rail side, and angle tan 0.05 on the Vee, these angles being connected, theoretically, by a curve of 1 300 - m. (65 chains) radius.

Running on the left-hand road. — Speed 100 km. (62 miles) p. h. — Switch taken facing.

At the first trial the superelevation at

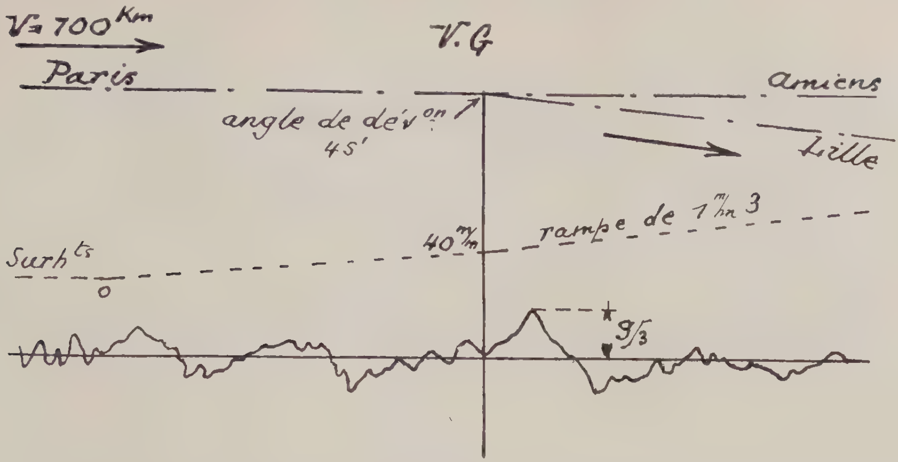

Fig. 5.
Note. — Rampe de... = rising gradient of...

the toe of the switch was 40 mm. (1 9/16 in.) (Fig. 5). The rearmost vehicle in which the Mauzin-Langevin quartz piezo-electric accelerometer was placed, was a coach of wooden construction, with bogies 13.68 m. (44 fr. 10 in.) apart, and weighing 40 tons. The accelerometer was stationed over the rear bogie.

On the straight portion, the superelevation at the toe of the switch tongue being considerably less (40 mm. instead of 90 mm.) than in the turnout with circular transition, and the cant gradient in front of the switches being correspondingly shorter, the vehicle did not cling to the low rail. On 5.50 m. (18 ft. 1/2 in.) beyond the toe of the switch (Fig. 5), the acceleration varies, as in the case of the switch of the circular turnout curve, from 0 to scarcely $g/3$.

This is explained by the fact that the switch angles of the two types of switches are practically the same, their tangents differing by only 2 mm. per metre (1 in 500). But for several metres beyond, the parabolic transition practically coincides with a straight line, the shock of entry into the curve does not occur, while the acceleration diminishes progressively over 5.50 m. (18 ft. 1/2 in.), becoming negative in that portion of the turnout curve where the superelevation is in excess of that required for the curvature on account of the speed. The vehicle then travels freely, as if on straight track, as far as the crossing.

This test gives us, with sufficient accuracy, the centrifugal acceleration due to a switch angle of 0° 45' to 0° 51', situated in track with 40 mm. (1 9/16 in.) of superelevation.

The second test was made with a superelevation of 59 mm. (2 21/64 in.) at the toe of the switch. The acceleration diminished appreciably and became $g/4$ at the switch (Fig. 6). Elsewhere the general form of the graph remained the same. The variation of the acceleration was not more than $g/4$ in 1/7th second, or $\frac{7g}{4}$ per second.

Now, in the turnout with circular curve the change of acceleration was $22g$ per sec.; the jolt was thus approximately 13 times less powerful in the case of the turnout with parabolic transition.

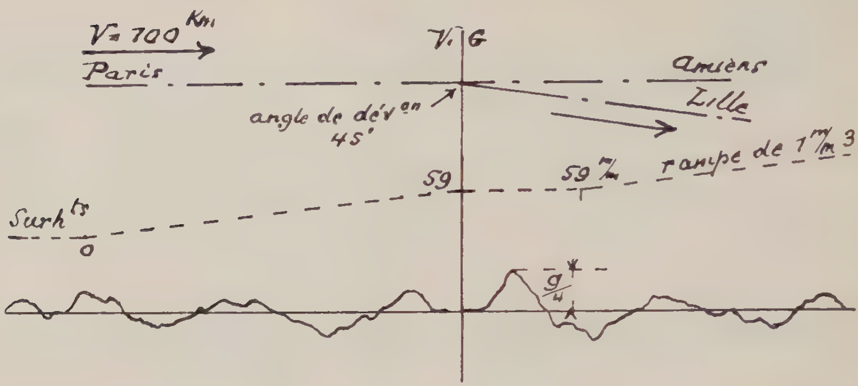

Fig. 6.

Running on the left-hand road. — Speed 110 and 120 km. (68.3 and 74.6 miles) p. h.

At 110 and at 120 km. the acceleration curves were very little different (Figs. 7 and 8).

The transit of the switch angle took place as already described although the variation is a little more rapid and more irregular than at a speed of 100 km. p. h. The acceleration rises to $g/3$ at 5 m. (16 ft. 5 in.) from the extremity of the switch tongue, and then decreases at an average rate of $2g$ per second.

Running on the right-hand road. — Speed 110 km. (68.3 miles) p. h. — Switch taken trailing.

The first test was made with a super-elevation of 30 mm. (1 3/16 in.) at the toe of the switch. The rearmost vehicle, in which the Mauzin-Langevin piezo-electric quartz accelerometer was placed, was a coach of wooden construction, with bogies 13.68 m. (44 ft. 10 in.) apart, and weighing 40 tons. The accelerometer was stationed over the rear bogie.

On the parabolic transition curve, the vehicle travelled practically as on straight

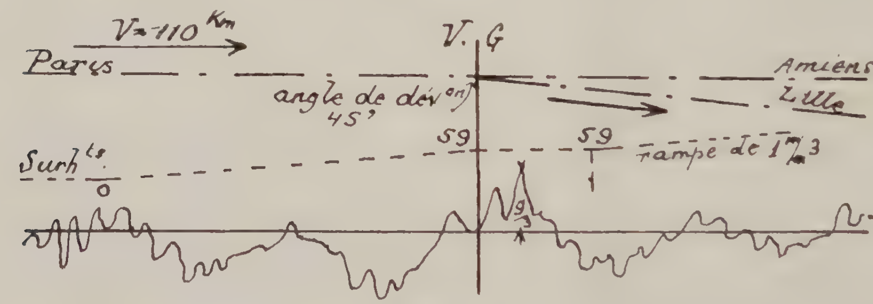

Fig. 7.

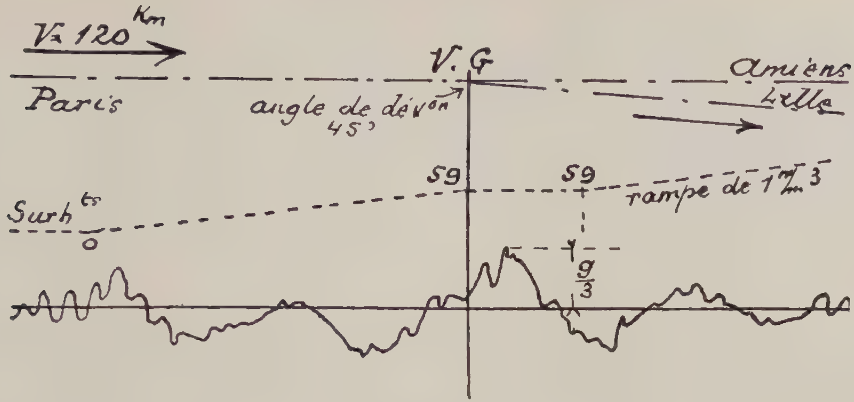

Fig. 8.

track (Fig. 9); on traversing the switch-angle, at 7 m. (22 ft. 11 5/8 in.) from the switch tongue toe, towards Paris, all the lateral play between the outer rail and the various parts of the vehicle being taken up, as in the case of the turnout with circular curve, the vehicle first strikes the outer rail with a force proportional to $g/2$, then rebounds towards the inner low rail which is struck with a force proportional to $g/3$.

The hunting motion thus set up rapidly dies away. Consequently from the point of view of the magnitude of the accelerations, the improvement obtained over the turnout with circular curve is negligible. But at the toe of the switch tongue, where the parabolic transition curve has ended, the suspended coach body is no longer acted upon by centrifugal force, and further, the superelevation is only 30 mm. (1 3/16 in.); con-

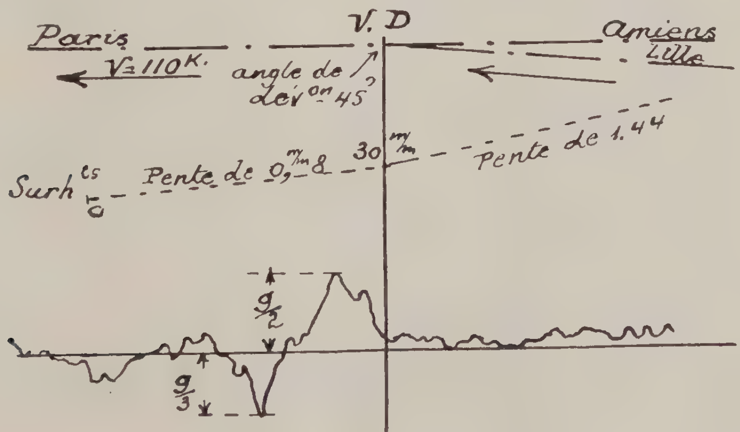

Fig. 9.

Note. — Pente de... = falling gradient of...

sequently the coach no longer acquires the rolling motion which it had with the circular turnout curve, and, as the extract from the Hallade record shows, the oscillation of the pendulum is no more than 27 mm. (1 1/16 in.) in amplitude, instead of 60 mm. (2 3/8 in.) (Fig. 10).

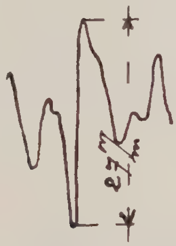

Fig. 10.

For the second test, the superelevation was increased from 30 mm. (1 3/16 in.) to 52 mm. (2 1/16 in.) at the toe of the switch.

After having traversed the switch angle, the vehicle acquires the hunting motion previously noted (Fig. 11).

The initial acceleration, corresponding to the blow on the outer rail, reaches $\frac{2g}{5}$, while the second acceleration, measuring the blow on the low rail is equal to $g/2$. The increase in superelevation does not therefore bring about any improvement.

As to the Hallade record, this resembles the one seen in Fig. 10.

For this test and for the following one, the rearmost vehicle, in which the piezoelectric quartz accelerometer was placed, was a coach of wooden construction, with bogies 12.63 m. (41 ft. 5 in.) apart, and weighing 34 tons, which is a little different from the preceding one. Notwithstanding this difference, we consider that the results obtained are comparable with those obtained with the 40-ton coach.

By comparing the results of the tests carried out on the right-hand road at a speed of 110 km. for the turnout with circular branching-off curve and that with a parabolic transition curve, we find that with the latter the rolling motion is very slight, the hunting motion remains, but the second blow is somewhat less severe, since the acceleration corresponding thereto is equal to $g/3$ instead of $g/2$.

To sum up : at speeds of 100 and 110 km. (62 and 68.3 miles) p. h., the turn-

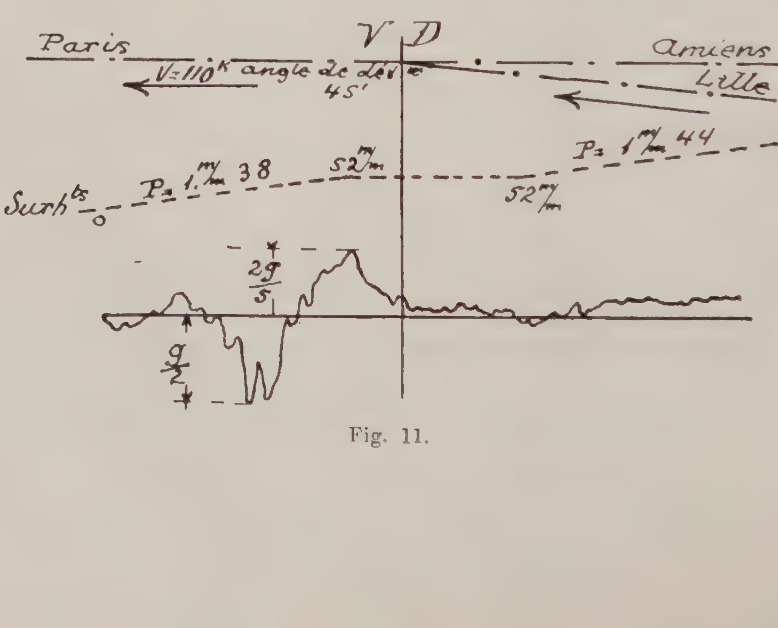

Fig. 11.

out with a parabolic branching-off curve, on the left-hand road, is clearly superior to that with a circular curve; the jolt at entry in the facing direction is 13 times less severe. On the right-hand road the accelerations at the exit from the switches have the same magnitudes but the rolling motion has become quite reasonable.

Running on the right-hand road. — Speed 120 km. (74.6 miles) p. h.

The superelevation amounted to 52 mm. (2 1/16 in.) at the toe of the switch tongue. The passage of the switch angle, in the trailing direction, takes place in the manner already described (Fig. 12). The acceleration corresponding to the blow on the outer rail is $g/2$, its variation and actual magnitude being comparable with the results obtained at the speed of 110 km. p. h.

The acceleration corresponding to the blow on the low rail amounts to $\frac{2g}{3}$. This value is greater than that for 110 km. p. h. when it only reached $g/2$, and the rate of change is a little more rapid.

The rolling motion seemed to be no greater than at the speed of 110 km. p. h., the Hallade pendulum record being almost identical with that obtained at the latter speed.

Improvements to be made to the layout of turnouts with parabolic transition.

The tests we have just described and discussed, demonstrate : (1) That the switch angle of 0° 45' 0", for a given superelevation and track gauge, and for a vehicle with a given type of suspension and degree of bogie play, gives rise to an acceleration of $g/4$ at a speed of 100 km. (62 miles) p. h., and $g/3$ at a speed of 120 km. (74.6 miles) p. h., the switch being traversed in the facing direction. (2) That at about 12 m. (39 ft. 4 1/2 in.) beyond the vertex of the switch angle the acceleration fluctuates about zero throughout the parabolic transition curve.

The effect of the switch angle is therefore isolated and consequently likely to be particularly noticeable. At the same time, over the rear wheels of the vehicle

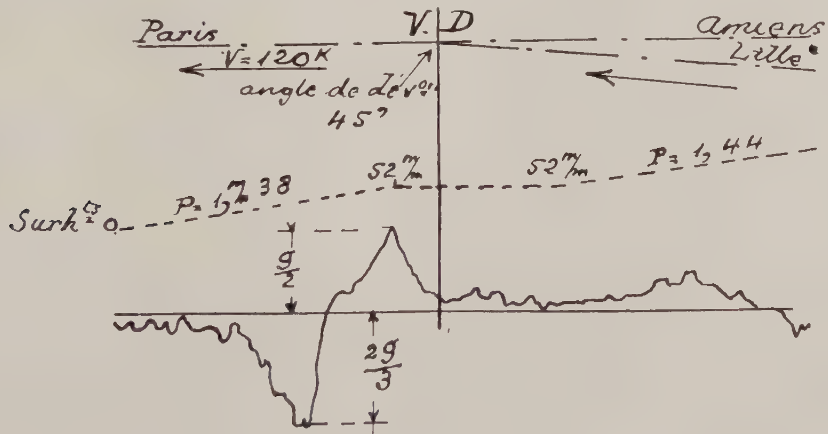

Fig. 12.

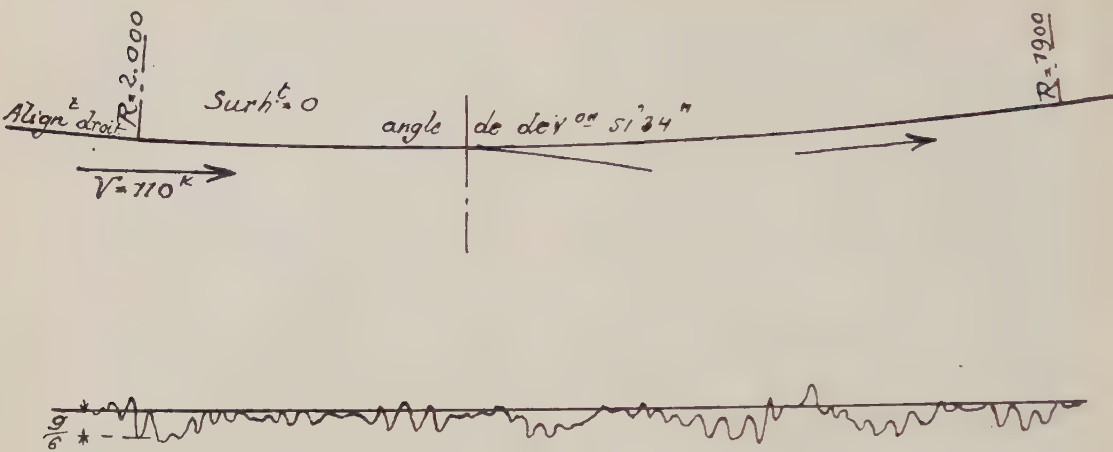

Fig. 13.

attached to the rear of the train, at a speed of 100 km. p. h. it is hardly perceptible, while at 120 km. p. h. its influence is no longer quite negligible. In a coach in the middle of the train, no effect at these two speeds can be felt by a passenger standing in the corridor. Nevertheless there is no doubt that a reduction of the switch angle is worth while.

To put such a reduction into effect means increasing the length of the turnout, as well as the length of the switch tongues, and there may be objections to increasing the latter beyond 15 m. (49 ft. 2 1/2 in.). Since the passage of the switch angle will always set up an acceleration, it seems useless to make the turnout curve immediately following the switch angle, of infinite radius, the acceleration for which is zero.

From the dynamic point of view it seems preferable to adopt for the turnout a transition curve the curvature of which at the origin generates a centrifugal acceleration (including that from the shock of entry into the curve) equal to that due to the switch angle. If the switch angle and the transition curve are suitably matched, a slight but uniform acceleration occurs for an appreciable distance along the turnout curve, instead of having at a single point on the turnout curve an isolated, and therefore more noticeable, maximum acceleration.

When travelling at 110 km. p. h. over a curve of 2 000 m. (100 chains) radius [1], without parabolic transition or superelevation, we find that at the tangent point, the shock of entry into the curve is equal to $g/6$ approximately (Fig. 13).

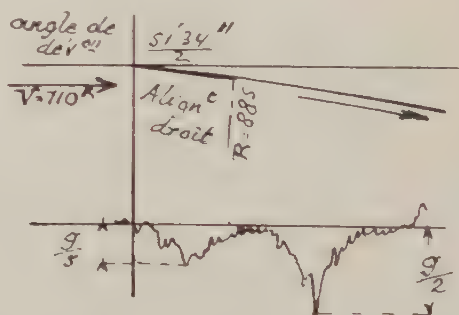

Fig. 14.

(1) The accelerometers were placed over the rear bogie of the rearmost vehicle, which was of wooden construction, with bogies 13.68 m. (44 ft. 10 in.) apart, weighing 40 tons.

On the other hand, the acceleration produced by travelling at the same speed through a switch-angle of 0° 25' 0" laid without superelevation, is equal to $g/5$ approximately (Fig. 14).

One can therefore tentatively adopt 1/3 000 as the curvature at the commencement of the parabolic transition; this allows a certain margin to cover possible local reductions of the radius of curvature, by distortion, immediately beyond the switch angle.

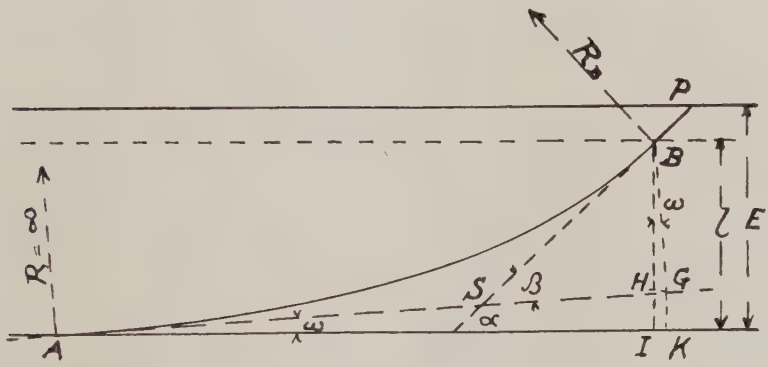

Fig. 15.

The foregoing considerations led us, in 1938, to suggest a new double-track junction having a turnout curve conforming to a parabolic transition, Fig. 16, the calculations for which are given at the end of this article (See Note 2). We believe that this type of turnout will permit of travelling through the branching-off roads at a speed of 120 km. (74.6 miles) or even 140 km. (87 miles) p. h. without appreciable jolts.

Turnouts with parabolic transitions, may also be laid in the following manner :

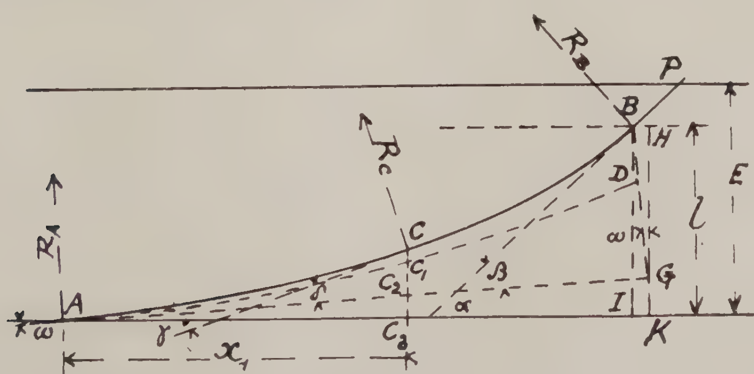

Fig. 16.

The turnout follows a curve of radius R, equal at the most, to the radius R_1 of the exit from the transition. At the same time the through road is no longer a straight line but a parabolic transition, having a curvature of $\frac{1}{R}$ at the switch toe, and $\frac{1}{R} - \frac{1}{R_1}$ at its extremity.

This alignment greatly simplifies the study of junction layouts in certain cases.

Superelevation.

Turnouts with parabolic transitions possess one unquestionable advantage over turnouts with circular branching-off roads, as regards providing the correct amount of superelevation with which the junction is to be laid. In turnouts with parabolic transitions, the superelevation can be applied progressively, as a function of the turnout curvature, whereas in turnouts with a circular branch track it cannot be other than constant throughout their length.

As a result of this, in the straight tracks at the switch toes A and C of the turnouts with circular branch curves, Fig. 1, the superelevation is as much as 8 or 9 cm. (3 1/8 to 3 1/2 in.) approximately, whereas it should be zero.

Although this may cause no serious inconvenience at the switch toe A of the turnout in the left-hand road, at the switch toe C of the turnout in the right-hand road the superelevation assists, as we have seen, the violent rebound of the vehicle on to the low rail, as the result of the change of direction imparted by the switch-angle.

In turnouts with parabolic transitions this drawback does not occur. Even though three or four centimetres (1 3/16 to 1 9/16 in.) of superelevation may be advisable at A, in order to mitigate the centrifugal acceleration due to the switch angle, the superelevation at C may be zero.

* * *

(2) High speeds in one direction only.

In cases where the directions MN and PQ are traversed at low speeds — 40 km. (25 miles) p. h. for example — and only the directions AK and CI are travelled at high speeds (Fig. 1), an excellent alignment for the junction is as follows : The sections of track HK and FI being laid as a rule on curves, the longest possible parabolic transition osculatory to the straight and the curve is introduced between each of these curves and the straight portions MN and PQ respectively.

The turnouts with circular branching-off curves AE, CF, and the diamond crossing GH are laid in such a way that their through roads lie in the transition curve [1], the switch toes A and C being at the commencement of each of the parabolic transitions.

The lead crossings and those of the diamond crossing (assuming various angles for the latter) are carefully chosen so that the branch roads of the turnout, and the opposite road of the diamond crossing afford an alignment suitable for a speed of, say, 40 km. (25 miles) p. h.

In existing tracks, a junction can be improved in this way when the turnout and the diamond crossing become due for renewal on account of wear. A ver-

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(1) « Courbes de chemins de fer, raccordements paraboliques (Etudes sans calcul intégral) » *(Railway Curves and Transitions* [without Calculus]), etc. Librairie de l'Enseignement technique, Eyrolles, publisher, Paris.

sines diagram of the alignment AEGH, or AFI, and the curve which follows on, enables a suitable parabolic transition, requiring the minimum slues, to be selected.

When the portions of track MA and PC may be curved, another solution consists in giving them the same radius as the branching-off roads AE, CF; or, in making them conform to the transition curve adopted. As in the previous case, the turnouts are laid so that their through roads follow the curvature of the branch, while the latter straightens out following an elongated sinusoid, or practically a straight line.

In existing tracks, either of the two solutions we have just described can sometimes be put into effect by mere slueing and partial re-screwing of the existing switches and crossings.

It can then be objected that the negotiation of the switch angles by the vehicles causes shocks, but tests have shown that at 110 km. p. h. a switch angle of 0° 51' 34" (Fig. 13) situated in a curve of 2 000 m. (100 chains) radius, without superelevation, does not give rise to any perceptible variation of acceleration.

These two solutions enable the superelevation to be applied rationally, as in the case of turnouts with parabolic transitions.

Office work.

Alterations to the alignment of junctions in curved roads are drafted by means of plans drawn to a scale of 1 in 200.

In spite of the care taken in preparing such plans, setting out on the ground often reveals discrepancies. We are of the opinion that the following procedure is to be preferred :

Investigate the possible solutions on a 1 in 200 plan. Prepare the versines diagram of the curves in which the existing junction lies. Construct the versines diagram (1) of the new curves for the junction which has been drawn on the 1 in 200 plan. Calculate the slues required at each of the pegs of the existing curves, in order to bring them into the new alignment.

It will often be found that slues which ought to be zero have relatively important values. These slues are eliminated in the process of adjusting the versines of the new alignment, the versines of the through roads and the turnouts, and also of the diamond crossing thus having predetermined values.

Pegging out on the site is effected in a simple and precise way by moving the pegs of the existing alignment by the amounts indicated in the statement of slues.

* * *

Note 1.

Let ABP (Fig. 15) be the outer rail of a transitioned turnout curve, consisting of the parabolic transition AB, osculatory at A (point of tongue) with the straight section AG, the latter forming the switch angle ω with the rail AK; P is the theoretical intersection of the crossing of angle α, B is the extremity of the wing rail.

We propose to calculate :

Firstly, the length AI of the turnout, from the toe of switch tongue A to the extremity, B, of the wing-rail.

Secondly, the radius R_B of the osculatory circle at B of the parabolic transition AB.

(1) For calculation of these versines, see « Diagrams of versines and superelevations », *Bulletin of the International Railway Congress Association*, October 1930. « Raccordements paraboliques (Etude sans calcul intégral) ». Eyrolles, publisher, Paris.

Calculation of [illegible]. — We have: [illegible]

[illegible]

[illegible], whence [illegible]

[illegible]

[illegible] : track gauge;

[illegible] : Length depending on the design of the crossing.

Calculation of [illegible]. — The equation of the transition curve is

[illegible], consequently: —

[illegible]

[illegible]

For $\alpha =$ [illegible], we find: [illegible]

Note [illegible]

Let [illegible] (Fig. [illegible]) be the inner rail of a turnout with [illegible] transition, following the transition curve [illegible] [illegible]

[illegible]

[illegible]

[illegible] the radius [illegible] of the [illegible]

[illegible]

[illegible]

[illegible]

Calculation of the ordinate CC_3 of the point C distant x_1 from the origin. —

We have :
$$y_c = C_2 C_1 + C C_1 = \frac{x_1^2}{2 R_A} + \frac{K x_1^3}{3 a^3},$$

$$C C_3 = y_c + C_2 C_3 = \frac{x_1^2}{2 R_A} + \frac{K x_1^3}{3 a^3} + x_1 \sin \omega.$$

Without serious error : $AC_3 = AC_2 = x_1$.

Calculation of γ. — We have :
$$\gamma = \delta + \omega.$$

From the equation, $y_c = \dfrac{x_1^2}{2 R_A} + \dfrac{K x_1^3}{3 a^3}$

we deduce that : $\tan \delta = \dfrac{dy}{dx} = \dfrac{x_1}{R_A} + \dfrac{K x_1^2}{a^3}$

Calculation of R_c. — By similar reasoning, we find :
$$R_c = \frac{a^3 R_A}{a^3 + 2 K R_A x_1}$$

For $R_A = 3\,000$, $\omega = 25'$, $\alpha = 2^\circ 36' 10''$, $E = 1.435$, $BP = 4.00$ we find :
$$AG = 54.44 \text{ m.}, \quad R_B = 935 \text{ m.}$$

We can determine all the dimensions of the transition curve AB in a different way.

We can produce this curve in the direction BA as far as the point *O* (not shown) where it is osculatory to a straight line (not shown) parallel to AK and below it.

In this way we obtain an ordinary parabolic transition, the curvature of which, at the origin, O, is zero, and $\dfrac{1}{R_A}$ at the point A.

The semi-overall length, *p*, of this transition is found to be given by the equation :
$$4 p^3 \tan \alpha - 6 p^2 l - \frac{(2 R_A \tan \omega)^3 \tan \alpha}{2} = 0.$$

By making use of the fact that $\tan \omega = \dfrac{x}{2 R_A}$ (where x is the distance between point A and point *o*), and that $\tan \alpha = \dfrac{p}{R_B}$, we are in a position to calculate all the dimensions of the transition curve AB.

[621. 132.8 (.73) & 621. 335 (.73)]

New American 5000-H.P. turbo-electric condensing locomotive for the Union Pacific Railroad,

by W. D. BEARCE,

Transportation Department, General Electric Company.

(*The Railway Gazette.*)

Nearly two years have been spent by General Electric and Union Pacific Railroad engineers in designing and building a 5 000-H.P. turbo-electric locomotive for handling fast and heavy passenger trains. This locomotive, which is now completed, is to be used on the fast and heavy express passenger services between Chicago and the Pacific Coast which have to negotiate 2.2 per cent. grades without a banking engine. Climatic conditions encountered *en route* range from — 40° F. in winter to 115° F. in summer. Some of the mountain passes through which the train must pass are over 8 000 ft. above sea level.

While the turbo-electric locomotive as a whole is still regarded by most engineers as experimental, the various pieces of apparatus out of which it is constructed have, for the most part, been thoroughly tried out and have demonstrated their reliability in other spheres. The assembly and arrangement of the equipment of this locomotive follow, in general, the practice in modern high-efficiency power plant work. Owing to the necessity for light weight and to limitations of space, some use has been made of experience gained in the installation of equipment on shipboard.

The boiler equipment differs from usual central station practice in the substitution of forced for natural circulation of the water through the tubes surrounding the furnace. The steam generated passes to a separator, where the excess water is removed by centrifugal action and then drained to the hot well. From the separator the steam goes through the superheaters and thence to the turbines. The exhaust steam goes to the condensers and the water returns to the hot well. The construction of this type of boiler precludes the possibility of any dangerous rupture due to any cause.

The electric motors, generators and control do not differ materially in design and construction from similar equipment supplied to trains now running. Briefly, then, the chief problem is the co-ordination of the several pieces of apparatus which go to make up the complete unit.

The operating advantages claimed for the steam-electric locomotive are as follow :

(*a*) Thermal efficiency from fuel to the driving wheels more than double that of the conventional steam locomotive.

(*b*) Electric braking resulting in saving in brake shoes and tyres, not only for the locomotive, but for the entire train.

(*c*) High rates of acceleration and braking due to high adhesive weight.

(*d*) Capacity for 500- to 700-mile performance without stops for fuel or water.

(*e*) Elimination of corrosion and boiler scale due to use of distilled water in a closed system.

(*f*) Elimination of unbalanced reciprocating parts which set up destructive forces in the rails, road bed, and supporting structures.

(*g*) Greater availability due to the construction of the boiler and absence of reciprocating parts.

Construction of mechanical portion.

The locomotive consists of two identical units capable of either multiple or independent operation under the control of one driver. Each unit consists of a 2-C-C-2 running gear surmounted by a single-ended streamlined cab.

The running gear of each unit consists of two three-axle driving trucks and two two-axle guiding trucks. The truck frames are integral nickel-vanadium steel castings. All wheels are of the solid type and all journals are in anti-friction bearings. The cab is mounted on centre plates, and the platform takes the buffing and pulling stresses. A flexible metal sleeve with sliding connection supplies ventilating air from centrifugal blowers located in the cab to each traction motor. The swivel truck arrangement provides room between the driving trucks for a well-type construction containing the steam boiler in the central part of the cab.

To secure smooth running at high speeds, restraint devices are used between the main trucks and cab, and between guiding trucks and main trucks. Side bearing pads on each main truck give the cab additional support.

The cab is designed to secure the lightest possible construction consistent with the requirements of strength and rigidity. The frame is built up with high-tensile steel tubular members, and aluminium cab sheets are employed except for the streamlined nose, which is of steel welded throughout. In the fabrication of the cab, welding is largely used, except for the aluminium sheets, which are riveted in place.

The locomotive is designed for single-end operation, with streamlining to minimise wind resistance. The front coupler is normally retracted and covered by a removable panel which conforms to the streamlined contour of the pilot structure.

Clasp brakes are used on both driving and idle wheels with four shoes per wheel on the drivers. The braking on these drivers is supplied from two cylinders per axle. A new high-speed air brake equipment suitable for use with both new and conventional trains has been included.

To facilitate maintenance, provision is made for replacing any unit of the power plant or electrical equipment, including the main boiler, in a few hours. Traction motors may be removed in a drop-pit. Other equipment can be removed by a crane through the roof.

Each cab encloses the following principal elements : a complete 2 500-H.P. geared turbine-driven generator set; a high pressure steam boiler; a compactly built turbine-driven auxiliary set with full automatic control; and a finned tube air-cooled condenser with turbine-driven fans for cooling.

The design of this locomotive is unusual in many respects, some of which are :

(*a*) The use of 1 500-lb. per sq. in., 920° F. steam.

(*b*) A type of boiler not previously used for railway service.

(*c*) Complete automatic control of boiler, auxiliaries, and power units.

(*d*) Provision for electric braking of sufficient capacity for holding trains on grades and for assistance in making service stops.

(*e*) Head-end auxiliary power for air-conditioning and other train electric service.

Many other features are incorporated in the design for the purpose of effect-

ing economies in operation, simplifying the duties of the operator, lengthening the cruising radius, and reducing maintenance cost.

This new Union Pacific locomotive is particularly adapted to lines where full electrification is not justified, but where superior passenger schedules are demanded. Each unit of this locomotive has the following ratings, weights, and dimensions :

Wheel arrangement per unit	2-C-C-2
Total weight (in British tonnage) with full tanks of fuel and water. . . .	236 tons 12 cwt.
Weight on driving wheels.	152 tons 13 cwt.
Weight per driving axle .	25 tons 9 cwt.
Fuel oil capacity	2 500 gall. (Imp.)
Water capacity.	3 330 gall. (Imp.)
Total length over couplers.	90 ft. 10 in.
Overall width	10 ft. 8 3/4 in.
Maximum height	15 ft. 0 3/4 in.
Maximum rigid wheel base	13 ft. 4 in.
Diameter of driving wheels	3 ft. 8 in.
Diameter of guiding wheels	3 ft. 0 in.
Rating of main turbines .	2 500 H.P.
Number and type of traction motors	6-GE-725
Gear ratio 65/31	2.097
Maximum operating speed.	125 m.p.h.

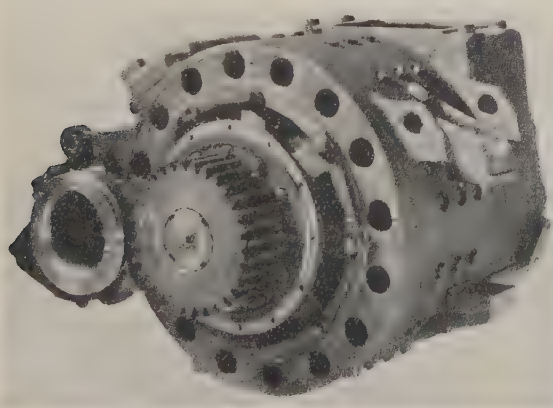
In order to obtain the capacity, flexibility, and efficiency essential to the best train operating characteristics, the power equipment has been designed to respond promptly to sudden demands for power. The rate of firing therefore increases and decreases automatically with the load demand.

Steam boiler.

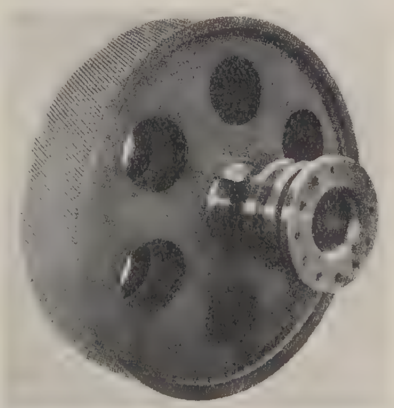
The steam boiler and automatic control equipment were designed and built by Babcock & Wilcox and the Bailey Meter Company, builders of boilers and steam control, in collaboration with General Electric. The boiler is a water-tube forced-circulation type, compactly built, incorporating a furnace, superheater, economiser, air pre-heater, and burners for Bunker C fuel oil. Special provisions are made to withstand shock and vibration resulting from the movement of the vehicle over the rails.

By replenishing water losses in the closed system with evaporator steam, practically all scaling and corrosion of the tubes are eliminated. The construction of the boiler unit and its 3-point supports furthermore avoids distortion of the tubes due to normal movement of the locomotive.

The economiser is an integral part of the boiler and utilises waste heat for increasing the temperature of the boiler feed water. Several boilers of this type have been built by Babcock & Wilcox



Nose-suspended d.c. traction motor.



Low-speed gear for main unit of locomotive.

View in cab showing train control fittings.

Driver's compartment.

and are successfully handling commercial service in stationary plants.

For starting the locomotive when cold a small vertical fire-tube boiler with a capacity of 100 lb. of steam per hr. is provided, using propane gas for fuel. This boiler supplies steam for heating the fuel oil and atomising the oil at the burners when starting the main boiler. The auxiliary boiler is designed for a pressure of 75 lb. per sq. in., and is used only for starting when an outside supply of steam is not available. Where steam can be secured at a roundhouse or from an external source the main boiler can be started without the use of this auxiliary.

Main turbine and auxiliaries.

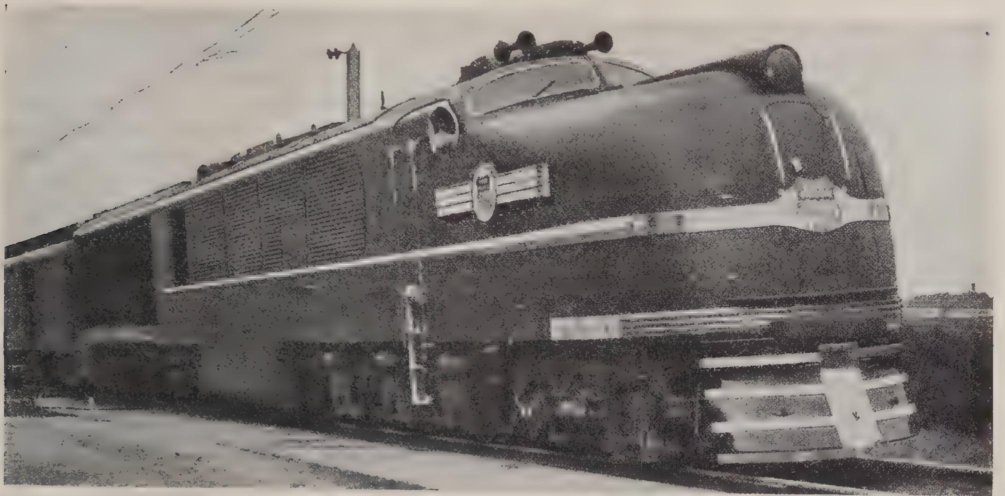
The main turbine generator set consists of the following elements :

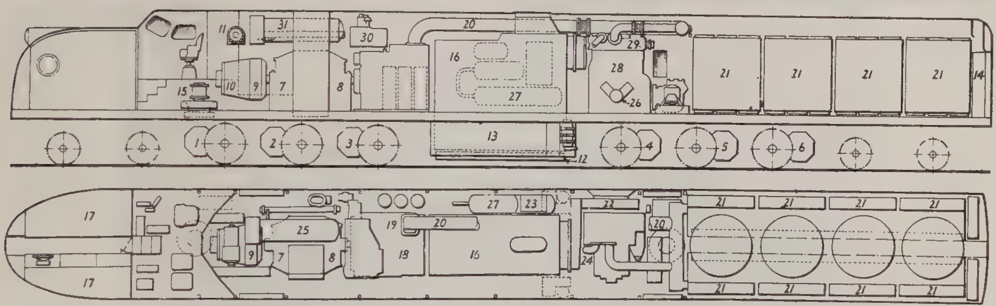
(*a*) High- and low-pressure turbines.

(*b*) A two-armature direct-current generator driven through a gear reduction approximating 10 to 1 from these turbines. This generator is self-ventilated from a fan located between the armatures, the air being drawn in through the commutator risers and discharged at the centre through the roof of the locomotive. In cold weather this warm air can be utilised for cab heating.

(*c*) A 220-volt 3-phase alternating-current generator connected to the main generator shaft through a flexible disc coupling. This alternator furnishes power for train air-conditioning, traction motor blowers, and other accessories.

(*d*) A variable voltage exciter, the armature of which is mounted on the same shaft as the alternator. This ma-





1-6 Traction motors.
7-8 Main generators.
9 Alternators.
10 Exciter.
11 Battery charging set.
12 Braking resistor.
13 Main control contactors.
14 Battery.
15 Traction motor blower.
16 Boiler.
17 Raw water tank.
18 High-pressure main turbine.
19 Low-pressure main turbine.
20 Exhaust header.
21 Air-cooled condensers.
22 Boiler draught fan.
23 High-level condensate tank.
24 Braking resistor separator.
25 Train heating evaporator.
26 Feed-water pump.
27 Feed-water heater.
28 Boiler auxiliary set turbine.
29 Condenser fan turbine.
30 Compressor.

chine supplies excitation for the main generator during motoring, and for the traction motors during electric braking.

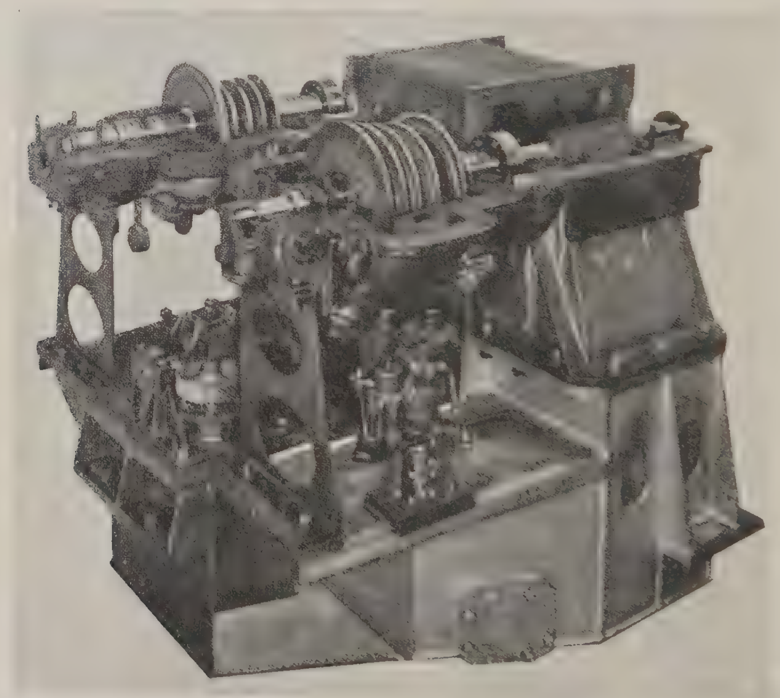
The auxiliary set is a variable-speed unit driven by a turbine which takes steam extracted from the main turbine. Its speed therefore varies somewhat with the traction load on the main turbine. This speed is further controlled automatically in accordance with the steam demand from the main boiler. Its function is to supply and regulate the combustion air and fuel oil delivered to the furnace, and also to supply feed water in proportion to the demand for steam. The complete set consists of a starting motor, an auxiliary turbine, combustion air fan, boiler feed pump, and fuel oil pump.

The lubricating pump for the auxiliary set is of the rotary-type, independently driven by a direct-connected 125-volt

direct-current motor. This pump is used for circulating lubricating oil to the reduction gearing, gear shaft bearings, turbine bearings, and feed water pump bearings. The circulating oil is cooled by radiators located with the condenser units. The condenser fan turbine is also independent of the auxiliary set drive, but is mounted on the same support. This turbine operates at a variable speed which is dependent on the condensation requirements but has a maximum speed of about 12 000 r.p.m. The condenser is mounted on each side of the rear end of the locomotive cab, and consists of finned-type vertical tubes. Headers at the top receive the exhaust steam from which the condensate is drained by gravity to a sump-tank under the locomotive cab. Ventilation for the condenser is provided by turbine-driven propeller-type fans drawing air through the sides of the locomotive and discharging it through openings in the roof.

As a part of the condenser equipment there is a steam-operated vacuum ejector for extracting small quantities of air which may leak into the closed system. This ejector normally will function under partial vacuum conditions down to 5 lb. per sq. in. absolute.

In normal operation the condensed water in the tank under the locomotive cab is maintained at a constant level by a float switch. Provision is also made for the addition of make-up water as required. Water from this tank is pumped into another tank located high in the cab, using a centrifugal type pump. In normal operation the upper tank will be kept full, the excess water overflowing and returning to the lower tank. An-

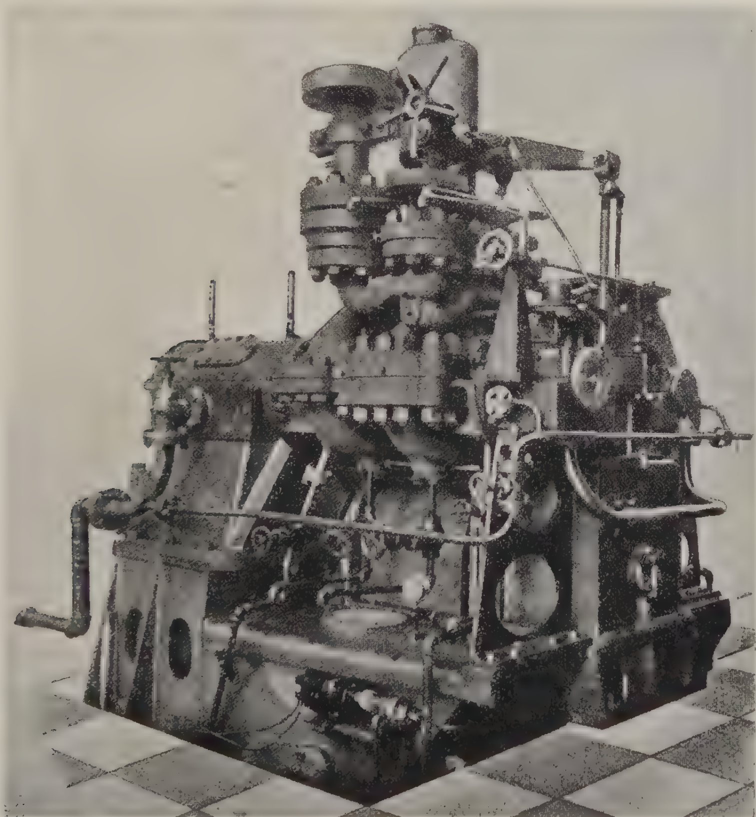


Steam turbine and gear with casing top removed.

other pump transfers the water from the upper tank to the suction side of the feed water pump. From this point the water passes through the feed water heater, the economiser, thence to the boiler tubes and then to a separator drum, from which excess water is returned to the sump. The steam passes through the superheater and turbines and back to the condensers.

A continuous supply of low-pressure steam is required for train heating, the operation of the air compressor turbine, and for heating fuel oil. For this purpose a heat exchanger or evaporator is used, consisting of a coil immersed in raw water. This coil takes steam either direct from the main boiler or by extraction from the main turbine. In either case the pressure is reduced before entering the coil. Water is supplied to the evaporator by three reciprocating pumps driven by direct-current motors.

A 150-cu. ft. double-stage air compressor designed to supply 125-lb. to 135-lb. pressure is driven by a steam turbine operating at 200-lb. pressure and driving through a reduction gear. This compressor set is regulated by a governor which operates a shut-off valve in the turbine supply, starting and stopping the set as required. A 15-cu. ft. compressor sup-



Oblique side view of complete unit.

plying air at 90-lb. pressure and operated by a 125-volt direct-current motor is provided for supplying air for operating the Bailey regulating devices and control equipment during starting when no steam is available for operating the turbine compressor set.

Electrical circuits.

This type of locomotive is essentially an electric locomotive carrying its own power plant. Since the main generator, however, is used solely for furnishing power to the traction motors, advantage is taken of the opportunity to regulate the train speed by varying the generator voltage, thus avoiding rheostatic losses. Acceleration of the train, therefore, is effected by controlling the current in the exciter field by means of the master controller, thus regulating the field current in the main generators. The main generator current is thus supplied to the axle-hung, geared traction motors at varying voltages, depending upon the demand for power and speed. Control current is supplied by a 125-volt motor-generator set with a battery floating on the line. Both acceleration and electric braking are regulated indirectly through the master controller.

This master controller includes an accelerating handle, an electric braking handle and a reverse handle. The reverse handle is also used as a selector handle for motor combinations in each direction. Provision is made for operating the motors in three combinations — series, series-parallel and parallel. Both the accelerating and braking handles normally hold a fixed kilowatt load on each controller step, except during the first few motoring steps, where approximately constant tractive effort increments are obtained.

The primary power for the auxiliaries is supplied by the a. c. generator. In addition to the operation of the two traction motor blowers this unit also supplies the motor-generator sets, supplying 125 and 64 volts d. c. Lighting circuits and headlights are also supplied from this a. c. source. Plug receptacles on the exciter of the locomotive at both sides permit obtaining alternating-current power from an outside supply. The 64-volt motor-generator set supplies power for the standard train circuits.

Automatic control features.

The power plant is entirely automatic in operation and there are no power plant pressure or temperature gauges in the operating cab. These devices are located on a control board situated in the apparatus cab. In the event of the operation of any of the protective devices, a warning gong rings in the apparatus cab and an indicating lamp shows at the driver's position.

Automatic train control and cab signal equipment of the continuous type with suitable inductors for operation over Union Pacific lines are installed on both units.

The mechanical and electrical parts of this locomotive were manufactured in the Erie plant of the General Electric Company, the boiler was supplied by the Barberton plant of the Babcock & Wilcox Company, and the boiler control devices by the Bailey Meter Company, of Cleveland. The locomotive is now undergoing tests, upon completion of which it will be placed in regular service by the Union Pacific Railroad.

62. (01]

Acoustic methods of sounding concrete and metal structures.

(From *Le Génie Civil.*)

Sounding large structures is a process of long standing which the engineer Rabut brought into common use when he developed the Manet-Rabut apparatus (which has been in general use some forty years) and applied it to the direct measurement of stresses produced by loads in motion on steel bridges on the French railways (1).

These measurements were of all the more value in that they showed that, owing to hyperstatic interactions which the designers had not allowed for in their calculations, these bridges were capable of carrying, either in their original condition or after judicious strengthening, the increased stresses due to the heavier and faster trains which were running over them. In this way the Railways avoided a considerable amount of very expensive reconstruction work.

The Manet-Rabut apparatus, now standard, are very good indeed, but require direct-reading, a fact which complicates and retards the measurements, especially in the case of large structures in which certain portions are not easy of access. In addition, these measurements only indicate deformations taking place during the short period covered by the test, since the apparatus cannot be left in place for a lengthy period without being thrown out of adjustment and otherwise damaged.

For some appreciable time consideration has been given to the design of apparatus making possible remote readings, and permanently installed without risk of getting out of order. There are various ways of effecting this, for theoretically each of the physical reactions produced by stresses in the medium under observation, or in the measuring apparatus attached to it, affords a means of solving the problem. For instance, a variation in stress can cause a variation in temperature or magnetic flux in some thermo-elastic or magneto-elastic apparatus; variations in pressure on a carbon cell or the junction of a bimetallic cell may produce a change in resistivity capable of measurement by instruments traversed by current flowing through the carbon or metal cell, etc.

Another variety of this type of apparatus is based on acoustics, and it is to this kind that Mr. Coyne, Chief Engineer for Roads and Bridges, has particularly devoted his attention since 1925 (1). With the co-operation of his colleagues and of well-known manufacturers he has evolved a method of applying this process, and he has expounded all the aspects of the question, which is of obvious importance, in a communication to the « Centre d'Etudes de l'Institut technique du Bâtiment et des Travaux publics ». We shall give a resumé of this, taken from the *Annales de l'Institut technique du Bâtiment*, which published

(1) M. Rabut published his « Recherches expérimentales sur la déformation des ponts métalliques » (Experimental investigations into the deformations of metal bridges), in the *Génie Civil* in 1892, 1893 and 1894.

(1) An account of Mr. Coyne's work in this sphere had already appeared in the *Génie Civil* for the 27th February, 1932, p. 225.

the description in its issue for July-August, 1938.

Principles of acoustic sounding.

A vibrating wire with its two ends attached to the object under observation is affected by its deformation, and hence its frequency of vibration varies with the expansions and contractions undergone by that object.

The wire is remotely excited by an electro-magnet in which flows a weak current (produced, say, by a small condenser) and the vibrations are detected on the same circuit by means of a valve amplifier at a central listening-in point.

At this central measuring point the frequency of the sound emitted by the wire has to be determined. To do this it is only necessary to compare it by means of the acoustic beats with the frequency of a standard wire (made of steel, as is the sounding wire) under the control of the operator, the tension of which he may vary at will. When the two wires are in unison, their tensions are equal, if they are of the same length, and proportional if they are of different lengths (1).

This is actually a matter of measuring tensions, or, rather, differences in tension between two successive conditions of the wire. As a matter of fact, it is unnecessary to know the zero position. All that is necessary is to measure the tension T for an initial condition and the tension T' for a new condition : the difference T' — T gives the stress during the interval. A positive reading indicates a tensile force, and a negative one a compressive force.

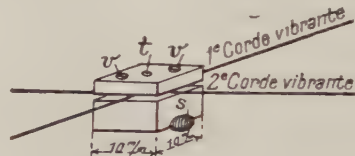


Fig. 1. — Method of securing acoustic wires to a metal part.

s, lower clamp block secured to surface by welding. — *t*, hole through both clamp blocks to facilitate crossing of the wires. — *v*, clamping screw for top block.

Note. — Corde vibrante = vibrating wire.

To calibrate the instruments at the observation post one or more controlling tuning forks enable the frequency of the standard wire to be compared with a known fixed frequency, so as to provide a method of zero correction, if such is necessary, at the moment of taking readings.

Such apparatus is extremely sensitive. In the laboratory it enables a variation in length of one micron per metre to be detected. Of course the sensitiveness is not so great in the field, where there are a number of adverse factors which reduce the accuracy of measurement to about 2 or 3 microns per metre, a figure which represents a stress of approximately 50 gr./mm² (71 lb./sq. in.) in steel and 0.5 kgr./cm² (7 lb./sq. in.) in concrete.

Method of construction.

When measurements have to be taken on steel parts the vibrating wire is gripped in screw clamps electrically welded to the part to be sounded. If the sounding affects a certain length, the number of fastening points of the wire is increased by spacing them about

(1) The formula for vibrating wires is :

$$N = K \frac{\sqrt{T}}{L}$$

where T = tension; L = length; K = coefficient depending on the specific weight of the wire.

Let : $N = K \frac{\sqrt{T}}{L}$ and $N' = K \frac{\sqrt{T'}}{L'}$ be the frequencies of the two wires. If N = N', we have :

$$\frac{\sqrt{T}}{L} = \frac{\sqrt{T'}}{L'} \text{ or } \frac{T}{T'} = \frac{L^2}{L'^2}.$$

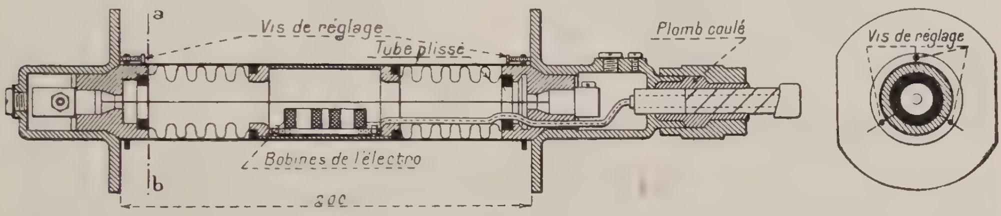
20 cm. (8 in.) apart. If a wide surface is to be observed several wires are used in the form of a squared network.

In the case of a concrete mass the wire is enclosed in a sealed tube embedded in the concrete, and the apparatus forms an « acoustic detector » (figs. 2 and 3) which permanently shows the conditions as regards tension or compression of the surrounding medium.

sound heard in the listening-in set is exactly the same as that generated by the detector.

An accurate preliminary calibration is not necessary, and even if the acoustic detector is somewhat mishandled while it is being placed in the concrete, the initial sound will alter, but will not affect the ultimate accuracy of the measurements.

The only possible errors are those



Section through *ab*.

Figs. 2 and 3. — Sections through acoustic detector for observations on concrete.

Explanation of French terms :

Vis de réglage = adjusting screw — Tube plissé = corrugated tube. — Bobines de l'électro = Magnet coils. Plomb coulé = lead seal.

The standard wire itself is stretched along the centre line of a steel tube : its tension is adjusted by a micrometer screw, and the instrument is termed a « frequency-meter ».

The listening-in set allows of the successive or simultaneous hearing of the acoustic detectors, the frequency-meter and the controlling tuning forks.

The advantage of acoustic apparatus as compared with the electric type, which soon becomes inaccurate, as has been shown by experience with large dams in the U. S. A., is primarily its simplicity, but especially the fact that it operates on the principle of *frequency measurement.* Dampness, bad contacts and defective insulation, which badly upset the working of electrical apparatus, have no effect on frequency. If the wire emits a sound and the electrical circuit is not open, the pitch of this

from thermal causes, if the apparatus and its support, although in thermal equilibrium, do not possess the same coefficient of expansion (case of an acoustic detector embedded in the concrete) : — When the temperature rises, the wire in the detector slackens, since it lengthens more rapidly than its sup-

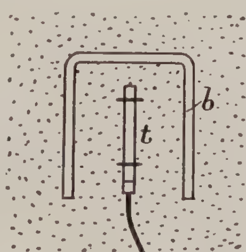


Fig. 4. — Arrangement of detector for temperature correction.

b, double-walled box. — *t*, detector shielded from surrounding stresses.

port, and the pitch of the sound drops : conversely, when cooled, the wire tightens and the pitch rises.

The simplest method of overcoming this source of error is to isolate one detector from the zone of stresses by enclosing it in a double-walled box (fig. 4); readings taken from it will show the error due to temperature differences, since it is only influenced by the latter; the error is then deducted from the readings obtained from the other detectors, so as to measure only the actual stresses to which the latter are subjected.

Laboratory tests.

Accoustic sounding may be employed in the laboratory in all cases where the parts to be examined are sufficiently large to allow of vibrating wires of 5 cm. (2 in.) minimum effective length to be fitted externally or internally. The sphere of application of the acoustic method is therefore very wide, especially when it is desired to observe deformations over a considerable period of time, and Mr. COYNE gives a detailed description of a series of tests he carried out at the Bridges and Highways Laboratory to demonstrate its usefulness.

We shall confine our remarks to a mention of the object of the tests : — determination of stresses existing in a mass of concrete; measurement of the contraction in a concrete beam, and of the elastic modulus of the concrete; action of the stirrups in ferro-concrete; measurement of the coefficient of expansion of concrete; gradual deformation (measured over a period of five years) of a reinforced concrete beam subjected to normal loads. This latter test shows up the lasting accuracy of the acoustic detectors when used in tests of long duration.

Applications to structures such as dams, quay walls, and penstocks.

Mr. COYNE then quotes a number of field tests, which show the ease with which the method may be applied in each case.

For instance, the measurement of the tensile stresses in ties supporting a quay wall constructed of metal sheet piles, at Havre, has made clear a number of facts which the author recounts; similar results were obtained in the case of the reinforced concrete ties of a quay wall at Dieppe (the acoustic detectors were placed in these ties during the placing of the concrete).

A particularly interesting example is the sounding of the dam on the Bromme, constructed in 1931 by the « Société des Forces motrices de la Truyère », and described in the *Génie Civil* of the 17th September, 1932.

In this arch dam, 40 m. (131 ft.) high, about twenty detectors were located in the contraction joints, and at the time of putting under load it was possible to obtain by this means a considerable amount of information of which Mr. COYNE points out the practical value.

Later on, the same process was applied on a much more extensive scale to the Marèges dam on the Dordogne, which was described in the *Génie Civil* for 26th October, 1935. About eighty detectors were located in this huge mass 90 m. (295 ft.) high, the majority being parallel to the outside facing and about 1.50 m. (4 ft. 11 in.) from it.

They are for the most part arranged in groups of three, forming an equilateral triangle with the base horizontal. This arrangement enables the directions of the principal stresses to be determined at each point under observation.

A certain number of these groups have a fourth detector in addition, which is relieved of all stress by locating it in a double-walled box, which allows a temperature correction to be made as described above.

By this means the deformations of the dam have been periodically observed

since the filling of the reservoir in June, 1935. These observations furnish most valuable information concerning the distribution of stresses in the structure, and it has been most satisfactory to detect each year the reappearance of the same stresses for the same conditions of loading and temperature.

Acoustic detectors have also been employed to control the hooping of the penstocks at the Marèges generating station. These are underground conduits of 4.40 m. (14 ft. 3 1/4 in.) internal diameter, made of concrete banded on the outside with cables of 70 mm. (2 3/4 in.) diameter.

Since the internal hydraulic pressure exceeds 100 kgr./cm² (1 422 lb./sq. in.) the cables, which form rings spaced 0.50 m. (19 11/16 in.) apart along the penstocks, were tightened up under a load of 135 tons so that the concrete conduit was subjected to an initial compression of more than 80 kgr./cm² (1 137 lb./sq. in.), and is thus capabe of standing up to the internal water pressure without difficulty.

To obtain the maximum degree of safety from this arrangement, the compression of the concrete must be kept under control, and the tension in the cables which maintain this compression must be watched continuously.

It has been found by means of the detectors that the sphere of action of one cable extends for more than 1 m. (3 ft. 3 3/8 in.) on either side, so that the concrete is maintained in a very uniform state of compression.

Lastly, acoustic detectors have been used at Marèges for keeping under observation the points of divergence of two metal penstocks feeding two secondary generator sets.

Finally, as an example of acoustic detection applied to bridges — either of metal or reinforced concrete construction — Mr. Coyne mentions the sounding of the Port-de-Pascau bridge, over the Garonne, in the Lot-et-Garonne Department. This is a three-span metal bridge with continuous lattice girders supporting a floor of reinforced concrete. In this case also acoustic detectors have enabled a study to be made of the stresses in the different members forming the lattice, and the part played by the reinforced concrete floor in relation to the main girders.

New applications of the method.

Apart from the cases just mentioned, acoustic detection may be employed for numerous measurements in the domain of physics, and Mr. Coyne examines two varieties of these : dynamic measurements and the measurement of subsoil pressures.

Dynamic measurements.

If the stresses vary on a time basis, it is only necessary to record the vibrations of the detector wire on a film by means of an oscillograph : the analysis of the record, which may be made afterwards at leisure, enables the curve of frequency variation — hence of stress variation — to be drawn to a time basis.

In practice, counting the oscillations recorded on the film is apt to be a lengthy and tedious process and hence may give rise to errors. When the stresses are not varying very rapidly, there is an easy method of simplifying this counting : it is only necessary for the oscillograph to receive simultaneously the current of varying frequency coming from the wire under observation and the current coming from a standard wire previously adjusted to a fixed frequency slightly different from the former.

In this way a record is made of the beats, which are themselves of varying frequency, and to obtain the required graph it is sufficient merely to count up the beats, which are far fewer than the primary vibrations.

This method has been employed on the metal penstocks of the Marèges gen-

erating station during one of the water-hammer acceptance tests made of the turbines. The turbine blades were closed in four seconds; their automatic closing was caused by the sudden interruption of the alternator electric circuit through the liquid resistance.

Measurement of subsoil pressures.

Exact knowledge of pressures existing in various subsoils is of outstanding importance to the engineer. A special arrangement of acoustic wires has been evolved for their measurement; it consists essentially of a diaphragm box comprising a rigid container closed at one end by a circular plate securely held all round its circumference. At each end of one diameter, the plate carries two lever arms, and between the ends of these the wire is stretched.

When the box is buried in the ground the plate is subjected to bending and the resulting angular displacements vary the tension in the wire. A noteworthy characteristic of the apparatus is that a deflection of a few microns at the most in the centre of the plate is all that is necessary to obtain adequate sensitivity for reading off purposes.

The same arrangement has been used to measure pressures exerted inside the mass of grain in a grain elevator bin, in proportion to the height of the mass of grain above the diaphragm.

Conclusions.

Mr. COYNE ends his account with the following summary of the advantages and scope of acoustic sounding :

Whether a laboratory analysis has to be made of little known phenomena, such as the contraction and gradual deformation of concrete, or wall effects, or if we have to verify the truth of rules used in construction; if in the structures themselves or in the subsoil we have to compare the calculated with the actual stresses — and when there is a discrepancy, as frequently happens — we have to amend theory in order to do better at less cost; if we have to observe the effects on structures of methods of force co-ordination, or to exercise instantaneous or continuous control over large structures, especially those most liable to failure, such as dams; finally, if we have to ascertain in due time, earth movements, especially the slow movements which cause the greatest mischief with structural work; then, the apparatus which you have before you will enable us to solve all these problems.

Nevertheless, many engineers hesitate to make use of it. Some object on grounds of expenditure. I have difficulty in believing that the few thousand francs required for a reasonable amount of observation work are of any account in comparison with the advantages of every description to be gained by a more exact knowledge of the stresses in the material, a knowledge from which we may gain a twofold benefit : safety and cheapness of construction.

Others object on grounds of time. It is true that the amount of time we can devote to scientific research is growing less and less. But the remedy is simple : there are specialised organisations capable of undertaking such observations and drawing conclusions from them.

Lastly, there is a third stumbling block. Some engineers are quite in the wrong as to what may be expected from this sounding method. They look for confirmation of their views. They would like the conclusions of theory, however outworn, never to be discredited. Actually, the reverse is the case. Measurements made with standard extensometers as well as with acoustic detectors are full of unforeseen results; and if it sometimes happens that the unforeseen result appears to arise from an error in measurement, in the majority of cases, it is mostly the measurement which is right and the theory which is wrong, because some consideration or other has not been taken into account in the assumptions made.

[625. 143.4 (.73) & 665. 882 (.73)]

New oxy-acetylene process for butt-welding rails.

(*Railway Engineering and Maintenance.*)

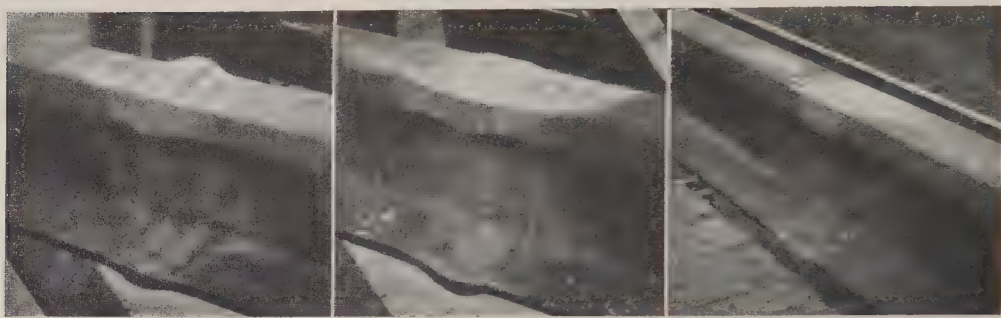
When the New Haven butt-welded rails into 800-ft. lengths for installation in four main tracks through its station area at Hartford, Conn., it utilized a new oxy-acetylene method in which, using special equipment, the abutting rail ends are brought together and heat and pressure are applied simultaneously until the weld is completed. Essential features of the new welding procedure and of the incidental operations involved and the equipment employed are described in this article.

General view of the welding operations showing (right to left) the welding machine, the oxy-acetylene cutting machine, and the annealing unit, to the left of which are the flat cars for storing the welded rails.

Utilizing a newly-developed oxy-acetylene method of butt-welding rails into continuous lengths, the New York, New Haven & Hartford recently installed eight lines of such rails, averaging 800 ft. in length, in the four main tracks through its passenger station area at Hartford, Conn. At this point, where the tracks are carried through the station area on a structural-steel viaduct, moving trains formerly occasioned considerable noise in the vicinity of the station, particularly in a passenger subway under the tracks, and it was largely for the purpose of reducing this noise that the continuous rails were installed. For this installation 112-lb. A.R.E.A. section rails were used, which were butt-welded into four lines having twenty-one 39-ft. rails each, and four lines having twenty-three 34-ft. rails each.

Essentials of process.

The welding procedure employed was developed and perfected by The Oxweld Railroad Service Company, a unit of Union Carbide and Carbon Corporation, and is known as the Oxweld automatic pressure rail-welding process. Using specially-designed equipment, this process involves essentially the uniform



Three stages of a butt-welded joint. Left — As it comes from the welding machine. Center — After part of the upset metal has been removed by the cutting blowpipes. Right — After the finish grinding has been completed.

heating of abutting rail ends to a temperature of about 2 280 deg. F., utilizing a mechanically-oscillated welding head that applies heat evenly to the rail sections from all directions. Simultaneously with the application of heat, the rail ends are forced together under a pressure which attains a maximum of 2 500 lb. per sq. in. Under these conditions, the rails are brought together in an upsetting action that involves shortening each rail 3/8 in. or each weld 3/4 in.

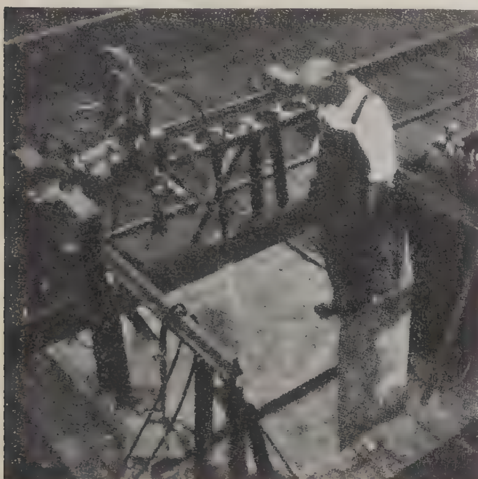
Another important phase of the procedure is the « normalizing » or stress-relieving operation to which the welds are subjected. In this process, the purpose of which is to achieve a refinement of the grain of the metal in the vicinity of the joint and to relieve internal stresses set up during the welding procedure, the joint is uniformly reheated to the critical temperature (about 1 380 deg. F.) and allowed to cool in the atmosphere. In this operation, the reheating of the joint is done with welding heads that are similar in design and arrangement to those used in the welding process.

Physical properties of welds.

To determine the physical properties of butt-welds made by this method, extensive laboratory tests have been conducted with specimens taken from welds made in 112-lb. rail. Representative results of these tests show that the metal at the butt-weld has a yield strength of 70 000 lb. per sq. in. and a tensile strength of 135 000 lb. per sq. in. with an elongation of 9 per cent. Tensile impact tests with specimens 0.236 in. in diameter show an elongation of 9.5 per cent under a load of 107 ft.-lb. In one series of tests it was reported that of a total of 20 specimens tested, 19 broke at points other than at the weld. In a rol-

The welding machine as seen from the receiving end.

ling-load fatigue test involving a welded rail supported as a cantilever, a load of 50 000 lb. was rolled back and forth across the joint through 2 000 000 cycles, and when failure did not occur at the end of this period the test was discontinued.



Removing upset metal from a newly-welded joint with the oxy-acetylene cutting machine.

In addition to the welding and normalizing operations, other steps in the butt-welding procedure include the grinding of the ends of the rail to a high degree of accuracy and cleanliness prior to the welding operation; removal of part of the upset metal by using a machine-guided oxy-acetylene cutting blowpipe; and the finishing of the joints by grinding. These and the welding and normalizing operations are performed in the following order : (1) Facing the rail ends; (2) making the butt weld; (3) removing the upset metal; (4) normalizing the weld; and (5) grinding the joint.

In the butt-welding work on the New Haven, all operations were performed on a string of flat cars of sufficient length to accommodate the equipment and the finished lines of rails, these cars being spotted on a conveniently situated yard track at Hartford. The layout involved the use of an adjacent track, from which the rails were transferred from cars to a rack car at the receiving end of the welding line. After the faces of the rails were ground, the rails were barred from the rack onto a conveyor line, consisting of rollers spaced at convenient intervals, by means of which they were conducted through the various operations of welding, cutting and heat treating.

Arrangement of equipment.

Equipment employed in the welding process included the welding machine, situated near the receiving end of the first car beyond the rack car; the oxy-acetylene cutting apparatus for removing the upset metal, which was placed a rail length away near the leaving end of the same car; and the normalizing unit which was located a rail length from the cutting operation near the center of the third or following car. All grinding work, including the facing of

In the normalizing operation the welds are reheated to the critical temperature (about 1 380° F.) and are then allowed to cool in the atmosphere.

the rail ends, was done with portable, power-operated grinding equipment at suitable locations on the cars.

Welding machine.

The essential features of the welding machine, which is mounted in a heavy frame of structural members, are the welding or heating heads, which are

Each length of rail was laid by pulling the string of cars from beneath it.

placed near the longitudinal center of the frame; the necessary parts for gripping the trailing end of the leaving rail and the forward end of the incoming rail; hydraulic equipment for applying the desired longitudinal pressure at the juncture of the two rails; and the necessary control devices. A heavy roller at each end of the frame facilitates the movement of the rails through the machine.

Equipment for holding the rails and applying the pressure includes two sets of grippers. The grippers at the forward end of the machine are stationary as to longitudinal movement and perform the function of holding the trailing end of the leaving rail in a fixed position during the welding procedure. The rear set of grippers, on the other hand, is free to move forward, or toward the stationary grippers, as the pressure and heat bring about the fusing and upsetting of the metal at the rail ends. Pressure is applied through the rear grippers by two hydraulic cylinders, one on each side of the rail.

Design of welding head.

Mounted between the two sets of gripper blocks is the welding head. Essentially the head is comprised of four tip blocks, placed above, below and on each side of the outline of the rail section, each of which contains a series of oxyacetylene tips. Thus, the abutting rail sections are practically encircled by tips in a plane perpendicular to the longitudinal center lines of the rails. In the tip blocks above and below the rail the arrangement of the tips corresponds in length to the width of the head and base of the rail, respectively, while the face of the block on each side is shaped to correspond roughly to the outline of the side of the rail section and contains tips that are directed toward the side of the head, the web and the upper surface of the base flange.

To permit the heating flames to be oscillated over the desired length of rail, the welding head is suspended from a track-mounted carriage that is operated back and forth longitudinally by means of a shaft from an oil cylinder. Both the length and rate of the oscillating motion are adjustable, and, on the New Haven job, the machine was normally adjusted

Upset metal on the sides of the head and the edges of the base flange at each welded joint was removed by a hand-held vertical wheel grinder.

to give a 3-in. movement of the welding head at a rate of 40 cycles per minute.

Control of the flow of oxygen and acetylene is effected by means of separate valves, including a quick-acting shutt-off valve for each tip block, which are mounted on a control panel together with the necessary gages. Auxiliary equipment at the welding machine includes a 5-H.P. 2-cyl. gasoline engine direct-connected to a two-stage vane-type pump which supplies oil to the hydraulic cylinders. This engine also operates a small pump for circulating cooling water through the tip blocks.

Removing upset metal.

When a weld has been completed, the line of rails is pulled forward a rail length by means of a hand winch located near the forward end so that the joints are in a position to permit the newly-made weld to be trimmed by the cutting blowpipe and the next previously-made weld to be normalized simultaneously with the welding of a new joint. In the trimming operation, use is made of a portable cutting machine, specially equipped to adapt it to the requirements of this operation. In brief, the unit consists of a motor-operated carriage on which are mounted two cutting blow-pipes with bevel-cutting attachments.

One of the cutting blowpipes on the carriage is adjusted to a horizontal position and is used to cut off the upset metal on the running surface of the rail in a horizontal plane. The other blowpipe is used to remove the upset metal from the sides of the head and the edges of the base flange in a vertical plane, and from the upper corners of the head on a bevel. The upset metal underneath the

The joints were surface-ground with a roller-carriage mounted cup wheel grinder which was tilted from side to side during the grinding operation.

base of the rail is not removed; when laying the rail provision for accommodating this metal when the joint comes on a tie is made by cutting a hole of the proper size in the tie plate with an oxy-acetylene blowpipe.

Normalizing machine.

In the normalizing unit, which is mounted in a light structural steel frame, the tip blocks, as stated previously, are similar in shape, design and arrangement to those in the welding machine. They are, moreover, suspended from a carriage having a movement parallel to the rail, which may be oscillated in the same manner as the welding head. In the case of the normalizing unit, however, the carriage is moved back and forth by means of a lever in the hands of the operator. A control panel similar to that in the welding unit also forms a part of the normalizing unit.

In the finish grinding of the joints the running surface and the sides of the head, and also the edges of the base, were ground to a smooth even surface. For the surface-grinding work a cup-wheel grinder in a roller-carriage mounting was used, the carriage being tilted from side to side during the operation in order to form the surface to the proper contour. For grinding the sides of the head and the edges of the base flange at each joint, a hand-held vertical-wheel grinder was used. In both cases the grinders were operated by means of flexible shafts from portable power plants of the « utility » type.

For facing the rail ends preliminary to the welding operations, a cup-wheel grinder on a special mounting was used. This device is fastened rigidly to the end of the rail by clamps and embodies a swinging arm that carries the cup wheel at its lower end. This wheel is mounted with the grinding face in a vertical plane at right angles to the rail and is provided with the necessary adjustments to secure a high degree of accuracy in facing the rail ends. The grinding wheel is operated through a flexible shaft from a portable power unit, also of the utility type, and is oscillated back and forth across the face of the rail end manually by the operator. Aside from the facing operation, other preliminary work done on the rail ends included the insertion of metal discs in the end bolt holes to prevent their distortion during the welding process.

As the butt-welding of each of the eight lines of rails progressed in the manner described above, it was moved along on the conveyor line which extended along the center of the string of cars for its entire length. For the storage of the welded rails pending their

One of the station tracks at Hartford after the continuous rails had been laid.

insertion in track, racks were provided on the cars on each side of the conveyor line and, as the welding of each line of rails was completed, it was barred onto one of these racks to make way on the rollers for the next line.

Gang organization.

For conducting all work incidental to the welding operation, an organization comprising 12 men was employed, including 2 grinder operators engaged in facing the rails; 2 operators at the welding machine; 1 man for operating the cutting equipment; 1 normalizer operator; 2 grinder operators engaged in the finish grinding and 4 laborers. With this organization, the average output was about 20 joints per day.

In unloading the rails at the point of insertion, the usual method was employed of anchoring the rail in the desired position longitudinally and pulling the cars out from under it. In laying each line of rails it was first barred back onto the rollers in the conveyor line and one end was fastened to the anchorage, which consisted in this case of several work-train cars with brakes applied. A heavy chain was used to fasten the rail, one end of which was connected to a clevis at the end of the rail while the other was looped around the coupler of the end anchor car. The cars were then pulled slowly out from under the rail and the free end of the latter was allowed to drop directly onto the track. In this manner the rails were unloaded without complications, in spite of the fact that the procedure required the laying of one end of each rail around a curve having a maximum curvature of 6 1/2 deg. The rails can, of course, be pulled off the cars if desired.

The rails were laid on single-shoulder five-hole tie plates and were fastened with compression clips at alternate ties, track spikes being used for lagging the tie plates to the ties. This type of construction was considered by engineers of the New Haven to be sufficient to restrain the rail from movement and for this reason no provision was made for expansion or contraction. For a distance of 100 ft. directly over the passenger subway further provision for reducing noise and vibration was made by placing a rectangular piece of 3/8-in. Fabreeka, a composition material with resilient qualities, under each tie plate to act as a cushion.

Observations made subsequent to the installation indicate that this measure has been successful in accomplishing the desired reduction of noise through the station area.

[621. 135.4]

The motion over curves of multi-wheeled locomotives,

by Chief Engineer AVENMARG, Munich.

(Glasers Annalen.)

For many years past designers of locomotives with many pairs of coupled wheels have endeavoured, by using radial wheel sets, to improve their running over curves, with the object of reducing the wear on the head of the rail and the wheel flanges. The most diverse solutions have been tried, the most important and oldest of which were undoubtedly those of Klose, Hagans and Klien-Lindner. Unfortunately they all had the drawback of decreasing the simplicity of steam locomotives, multiplying the number of details used in the construction of these engines, and resulting in increased maintenance work and cost.

Helmholtz and Gölsdorf were the first to retrieve the situation by simply giving the wheels a small side play in the axleboxes and coupling rods, and dispensing with any centering or similar device. The wheels guided themselves by their flanges coming into contact with the heads of the rails. The Class 170 2-8-0 locomotive designed by Gölsdorf 40 years ago, which had an Adams leading pair of wheels, the first pair of coupled wheels rigid, the second with side play, the driving wheels rigid and the fourth pair of coupled wheels also with side play, was a masterpiece from the point of view of running over curves and could not be constructed more rationally to-day. Without having a centering gear, the leading wheels govern the deflection of the locomotive, on entering a curve, by means of the oblique axlebox cheeks, so that the first rigid coupled pair of wheels has only to complete this movement and rotate round the driving axle. As, owing to the side play on the second and fourth coupled pairs of wheels, there is no additional pressure on the flanges, long periods elapse before the tyres are to be returned.

However, the 0-10-0 type locomotive, designed on the same principle and adopted by many railways, soon showed defects. The leading pair of wheels, which was for the first time a coupled pair with side play, guided itself well over the curves, but it left to the second coupled pair of rigid wheels the work of deflecting the whole mass of the locomotive. As this pair of wheels, placed near the centre of gravity, works on a small lever arm and its striking angle with the rail is relatively great, its flanges soon become sharp. This especially showed itself on the G. 10 locomotive of the former Prussian State Railways where unfavourable circumstances were further aggravated by the fact that to avoid long piston rods the centre pair of wheels was chosen as the driving pair. Here again the displacement was carried out by the second rigid coupled pair of wheels, wherely the pressure on the flanges was increased by about 50 %. This arrangement does not appear, however, to have been satisfactory, as this subject was dealt with in the *Mitteilungen* of the Central Office of Railways, Berlin, in 1911. In 1929 it was finally decided to make the fifth pair of coupled wheels rigid and to reduce the thickness of the flanges of the third and fifth pairs of wheels by 10 mm. (3/8 in.). In consequence the first coupled pair of wheels was left with a side play of 25 mm. (1 in.) in both directions, and

the second coupled pair of wheels had to guide the locomotive over the curves as previously. The locomotive works a little more freely over the curves and turnouts because two pairs of wheels have thin flanges; but there is little change as regards flange wear on the second pair of wheels. By making the first pair of wheels rigid and giving twice 25 mm. (1 in.) of play to the second and making the flanges of the third pair of wheels thinner, an improvement could have been anticipated. In this way the rigid wheelbase of 4.500 m. (14 ft. 9 3/16 in.) would be maintained and the prescribed tyre turning intervals would certainly have been increased. This arrangement of the wheels would then have corresponded to that of the Russian 10-coupled locomotive, large numbers of which type have been built by German Locomotive Works.

For tank locomotives, the running qualities of which must be equally good when running forward and in reverse direction, the unsymmetrical arrangement, if one can call it that, of side play of the various pairs of wheels is obviously not suitable. Helmholtz, therefore, logically developed tank locomotives with 4 pairs of coupled wheels, the end pairs of which are mounted rigid in the underframe, while the two central pairs can move transversely (Fig. 1).

Fig. 1. — 0-8-0 tank locomotive, the second and third pairs of couples wheels of which have side play.

The use of this arrangement, which produces equal wear on the flanges when working in either direction is, however, limited by the length of the connecting rod or, what comes to the same thing, the rigid wheelbase permitted, as in this case the fourth pair of coupled wheels becomes the driving pair.

In order to have also 10-wheeled tank locomotives with acceptable flange wear and as long as possible tyre turning intervals, the *Krauss-Maffei* Locomotive Works fitted a number of locomotives with a guiding arm, principally on locomotives for the mountain lines of Northern Spain. The first German tank locomotive with this arrangement was a 10-coupled engine put in service on the Brohltal line (Fig. 2).

Fig. 2. — 0-10-0 tank locomotive, with guiding arm between the first and second pairs of coupled wheels.

On this line, with 1 in 20 gradients and curves of 50 m. (2 1/2 ch.) radius, the result was most successful. The distance covered by the locomotive between two re-turnings of tyres reached 25 000 km. (15 500 miles) as against 8 000 km. (5 000 miles) on the other locomotives working on the same line. Thereupon 45 0-8-2 tank locomotives of the Bavarian system of the Reichbahn were constructed with this guiding arm between the first and second coupled pairs of wheels (Fig. 3). The fourth coupled pair of wheels forms with the fifth, which is the carrying pair, a Krauss-Helmholtz bogie. The driving

wheels are the only ones mounted rigidly in the underframe. Thus the engine is guided in each running direction by two pairs of wheels and runs over the curves without difficulty.

The last application was to a 10-coupled tank locomotive of the Kassel-Naumburg Railway (Fig. 4). This Railway Company had some locomotives with a first and fifth Gölsdorf pairs of wheels with side play already in service and wished to improve them by reducing the flange wear and make the running steadier at high speeds. The new locomotive was fitted, like that of the Brohltal line, with a guiding arm between the first and second pairs of coupled wheels, the third and fifth pairs mounted rigid in the underframe and a fourth pair with side play. Consequently, in both running directions two pairs of wheels guide themselves whilst the deflection of the locomotive when working forward will be looked after by the pivot of the arm and when in reverse by the fifth pair which is a long way from the centre of gravity of the system. These locomotives fulfil

Fig. 3. — 0-8-2 tank locomotive with guiding arm between the first and second pairs of coupled wheels.

Fig. 4. — 0-10-0 tank locomotive with guiding arm between the first and second pairs of coupled wheels.

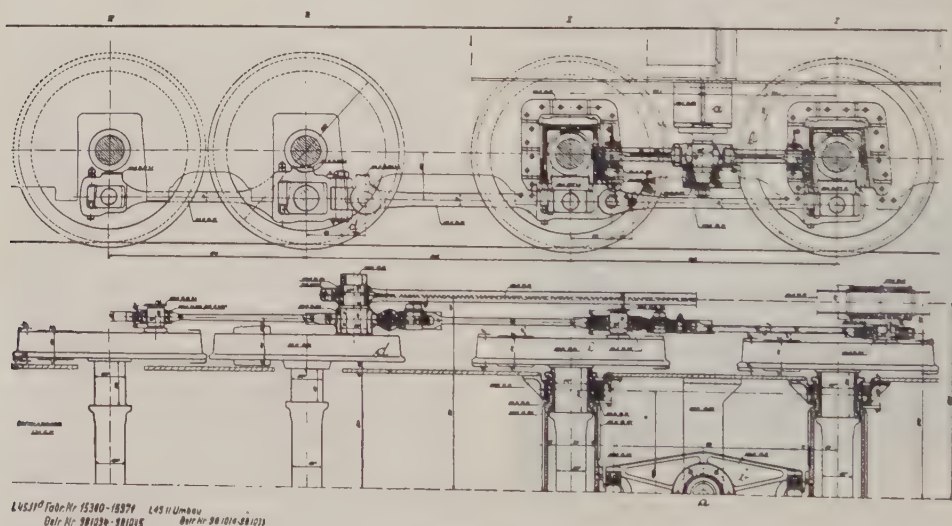


Fig. 5.

the essential condition for steady riding at high speeds, as the rigid wheel base between the third and fifth pairs of wheels is short so that they work over curves without difficulty, and have great guided length from the pivot of the guiding arm to the fifth pair of wheels.

The accuracy of this view was confirmed by the trial runs of this locomotive. It runs over the line with a steadiness greatly superior to that of the old locomotives of the same type fitted with end radial wheel sets, and the tyre returning intervals are longer. Figures 5 and 6 show the arrangement of the guiding arm.

The lever with equal arms *b* between the first and second pairs of coupled wheels, of which each has 2 × 20 mm. (13/16 in.) side play oscillates round a pivot *a* rigidly mounted in the locomotive underframe, and thus carries out the guiding of the locomotive when travelling forward. On a straight track this lever remains in its central position even at the highest speeds, but deflects immediately on entering curves, as has been proved by extensive trials. Centering or check springs are therefore unnecessary. The double lever itself does not need any special care in service, there being only two lubricating caps to feed. When the wheels are taken out, the lever remains on its spring support on the locomotive underframe, and when the wheels are put back again it engages with the driving cams without any detail having to be unfastened or removed. The unsprung weights are not increased by the lever *b*.

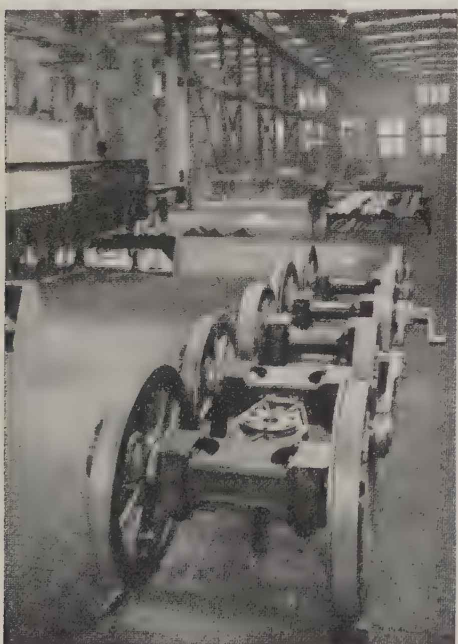


Fig. 6. — Wheel sets of 0-10-0 tank locomotive with guiding arm.

Naturally the coupling rods between the first, second and third pairs of wheels must follow the deflection of the wheels. In order to obtain the necessary mobility, a vertical pin *d* has been provided near the driving crank around which the two front coupling rods can oscillate. The front coupling pin carries with it the rods during the deflection of the wheels whilst the lengthened coupling pin *e* of the second axle is pushed back across the coupling rod bearing.

In order to compensate for the slight obliqueness of the coupling rods in relation to the coupling rods which move parallel (inclination approximately 1 in 150) the rod bearings are fitted with spherical heads, following the method of construction already used for the standard German State Railways locomotives, classes 43 and 44. The large coupling pin has to take up the slight lateral force which is produced and therefore receives a suitable shape.

The guiding arm preserves the flanges and, consequently, also the track; owing to the simplicity of its construction, it is a remarkable advance in the general arrangement of locomotives having many pairs of coupled wheels, intended for running easily over curves in both directions.

[656. 231 (.89)]

A simplified rate classification in Uruguay.

A new scheme which provides an automatic goods classification designed to place rail and road transport on a equality.

(*The Railway Gazette.*)

Owing to the serious encroachment of road competition and consequent loss of traffic, the Central Uruguay Railway decided, in July, 1936, to take drastic measures by introducing an entirely new system of classifying and rating merchandise, and there follows hereunder a description of the main aspects of this revolutionary scheme.

Outline of scheme.

In general terms, the scheme was designed to put the railway on an equal footing with lorries as regards rate charged and unification of load. The old system of several classifications based largely on the intrinsic value of commodities, each with a minimum charge (and the application of the highest rate where a mixed classification was concerned) left the railway at a grave disadvantage *vis-à-vis* the road haulier, to whom a ton is a ton, be it of silk stockings or fencing wire. Moreover, the system of ordinary goods tariffs designed on the usual tapering curves (in force prior to July, 1936) was fundamentally obsolete, as it pre-supposed monopoly of transport by a wide differentiation between « high » and « low » grade goods, and charges proportionally higher for short than for long haul. Added to this was the fact that, up to the time it was decided to make a change, the bulk of the railway's traffic was handled at special tariffs, of which no fewer than 8 000 were in existence, and this unfortunate necessity introduced anomalies inimical to traffic development. The new scheme also contemplates, as a primary factor, the value of services rendered, i.e., the actual cost of transport, not merely the value of the article transported.

First and foremost, therefore, the classification of commodities in various categories was entirely suppressed, and merchandise divided into two main classes consisting of : (*a*) wool, hides, and skins, and (*b*) all other articles; the separation from the remainder of the three traffics specified in (*a*) was based on their volume or weight ratio, justifying the application of slightly higher rates.

Very low rates were entirely eliminated, with the result that broken stone is now rated basically on an equality with general merchandise. To adjust theory, however, to practical fact, a sliding scale of charges was established varying in inverse ratio to the volume of the consignment, so that, with broken stone moving normally in consignments of 50 tons and above and grocery in consignments of from 1 to 10 tons, each automatically fell into its propre place in relation to the other, while the basic cost factor was retained in each case. This method is superior to that in which rebated rates were conceded in return for a tonnage contract, in that it contemplates the actual fundamental economics of railway operation — in other words, full wagon loads, full engine loads, and the reduction at terminal and intermediate stations of the handling of isolated wagons.

As a complementary measure new formulæ for rating were evolved from a

study of traffic density (ton-kilometres) and the characteristics of lorry competition in the various zones served by the railway. The practical result of this was to make the rate per ton-kilometre higher as the distance from the capital increases — *i.e.*, in the zones more remote from competition; to cite an example, a ton of traffic carried 25 km. in a zone near the capital now pays a considerable cheaper rate than it would for the same distance 300 km. further away, and the rates have been scaled in such a way that the sum of those corresponding to a number of stages is equal to that applying to the throughout distance.

Evolution of rating formulæ.

1. *Zone Factor « Z ».* — The construction of excellent roads parallel to the railway and the exploitation of water routes resulted in endowing large areas of the country with new « geographical privileges » in relation to other zones, and the rating system adopted had to be flexible enough to take these privileges into account, for a railway cannot apply high tariffs, however justifiable they may be economically, if a cheaper form of transport or a more direct route be available to traders. Therefore the zone conditions were assessed and expressed in terms of ratio, determined by the nature and routing of roads, prevalent gradients, etc., and relative level of lorry (or water) transport costs; by keeping to ratios, the figures could be made to hold good for large or small vehicles, owner-driven or company-operated, economically worked or extravagantly handled; the only premise is that, if in a given zone a lorry can operate at « X » cost, then its costs will be in the order of 20 per cent higher in a zone which bears to the former a ratio of 1.2 to 1. Road traction ratios were determined by an exhaustive series of surveys; these were plotted as curves, and by the incorporation of such considerations as arose from the relative distances by road and rail, and from any water route virtually short-circuiting the railway, were classified as « Goods tariff zone ratios ».

As a corollary of the foregoing, the zone dsitance factor formed of zone-factor « Z » multiplied by distance in kilometres « D » becomes, in effect, a compensated kilometrage-table, applicable strictly geographically, and possessing superiority over the old kilometric scale, which pre-supposed equality of zone conditions throughout the line.

2. *Commodity Index « I ».* — Reference has already been made to the abandonment of the old and complex goods classification and the concentration of merchandise in two broad categories : (*a*) wool, hides, skins, etc., and (*b*) all other goods. It should now be mentioned that (*b*) is subdivided, by admitting a lower zone distance factor for wheat, maize, rice, etc., which, because of their bulk movement and special economic characteristics, it is necessary to consider as a class apart. Reference has also been made to the sliding scale of charges varying in inverse ratio to the volume of the consignment. In effect, this means that the first ton has a higher rate than each ton of the subsequent five (or ten), which in turn is higher than each ton of the subsequent fifty, and involves a variation of the commodity index « I », covering a range from 3.1 to 1.9. The level of these indices has been determined by equating a known traffic (that of the previous year) in zone-ton-kilometres, so as to yield a given total revenue, and the relation of the indices one to another has been settled by the consideration of lorry practice. In short, the whole thing is the result of trial calculations, with a view to evolving a compromise between high figures (advantageous to competitors) and low figures (sacrificial of potential revenue).

3. *Terminal Factor « T ».* — All handling and terminal costs are regarded as constants, and added to freight calculated by formula. In order to give it flexibility, this factor has been designated as a constant per consignment, plus a constant per 100 kgr.; *e.g.*, the rate for a large consignment of fencing wire would bear a given relation to the cost of alternative transport, yet a consignment of 100 kgr. would pay more per ton, not only because of the sliding scale of indices, but also by the incidence of full per consignment terminal.

Map of Central Uruguay Railway showing zoning.

The following is a working expression of the basic formula of the new tariffs, without taking into consideration the effect of the variables : I × ZD + T = tariff per ton in local currency. Fully expressed — *i.e.*, taking the variables into account — the formula in its complete form would be :

For consignments not exceeding 1 000 kgr. :

$$K + WI\ (ZD) + nk = Tf.$$

For consignments exceeding 1 000 kgr. but not greater than 5 000 kgr. :

$$K + WI\ (ZD) + nk + W_1\ I_1\ (ZD) + n_1k = Tf.$$

For consignments of over 5 000 kgr. but not exceeding 50 000 kgr. :

$$K + WI\ (ZD) + nk + W_1I_1\ (ZD) + n_1k + W_2I_2\ (ZD) + n_2k = Tf.$$

where K is the fixed charge per consignment; k the terminal per unit of 100 kgr.; ZD the zone-distance ratio; W, W_1, W_2 the weight in kgr. respectively, between 1—1 000 kgr., 1 001—5 000 kgr., and 5 001—50 000 kgr.; I, I_1, I_2 the commodity indices, for the same range; n, n_1, n_2 the number of units of 100 kgr.; and finally Tf the total charge in pesos.

Values assigned to factors.

As a result of the zone surveys, the following zone factors (ZD) were adopted :

Z = 1	Central—25 de Agosto 25 de Agosto—San José Mal Abrigo—Colonia Rosario—Puerto Sauce Cardona—Mercedes Sayago—Pando	
Z = 0.8	Pando—Minas	A highly competitive zone served by a first class macadamised road, which also drains an extensive area beyond Minas.
Z = 1.05	San José—Mal Abrigo 25 de Agosto—Florida Toledo—Fray Marcos Retamosa—Treinta Tres	These are zones in which railway operating costs progressively increase due to diminishing density of traffic, and in which road competition is progressively less intense.
Z = 1.1	Florida—Durazno Mal Abrigo—Cardona	
Z = 1.15	Nico Perez—Retamosa	
Z = 1.25	Fray Marcos—Nico Perez	
Z = 1.3	Durazno—Paso de Los Toros	
Z = 1.5	Paso de los T.—Tacuarembó Nico Perez—Melo	In these zones proportionally higher rates can be applied to raise the general level of the average.
Z = 1.75	Tacuarembó—Rivera	

It should be noted that for wheat, maize, rice, etc., a separate series of ZD coefficients is applied, using unity as the zone factor, wherever the normal scale shows a higher reading. In order to comprehend more fully the system of zone assessment and rating, reference should be made to the map of the Central Uruguay Railway.

With regard to the goods indices (I), the following are the values assigned :

Wool, hides, skins, etc.		
Each ton up to 10 tons	In same consignment	2.9
Each ton above 10 tons		2.7
Other goods.		
First ton	In same consignment	2.6
Each ton of subsequent 4 tons .		2.3
Each ton of subsequent 45 tons .		2.1
Each ton of above 45 tons . .		1.9

Terminal factors (T) have been fixed at :

Per consignment	$0.25
Per 100 kg. or fraction . . .	$0.05

For publicity among clients of the railway and for employees' use a table was prepared for each station, embodying the ZD factors applicable between that station and all others on the system. Hence no calculation of any kind is necessary to obtain a coefficient from

which, by reference to a ready reckoner, the actual freights in currency are extended in the appropriate column corresponding to the respective item of the goods indices, leaving only to be added the terminals, which obviously cannot be reduced to a per ton basis.

In summarised form, the new scheme provides an automatic goods classification, spacing out, relatively widely, goods habitually handled in wagon or train loads from those normally moving in small lots.

The new rates do not apply to interchange traffics with Brazil, the Midland Railway group or the State Railways Eastern Line; but, with the elimination of these, figures illustrative of the success of the scheme show that, notwithstanding a heavy decrease in wheat tonnage, the traffics to which the new rates apply have so far given satisfactory increases, principally from previously low-rated commodities such as stone, and despite the considerable decrease in revenue due to lower rates from what was previously the highest rated traffics.

[625. 232 (.42)]

New buffet-restaurant car for York-Swindon service.

An important service operated by the London & North Eastern Railway Company is that which conveys through carriages from Aberdeen to Penzance and the South of England. The train leaves Aberdeen at 10.20 a. m. each week day, and by way of Edinburgh, York, Sheffield and Swindon, passes to the Great Western line.

The restaurant-car service is provided by the L. N. E. Company, and in order that facilities may be available for the provision of substantial meals or light refreshments, two new vehicles known as Buffet-Restaurant Cars have been specially built for this service.

The cars were constructed at the Doncaster and Dukinfield Works of the L. N. E. Company, to the designs of Sir Nigel Gresley, Chief Mechanical Engineer, to whom we are indebted for this information, and combine the facilities of a restaurant-car with those of a buffet car. There are two saloons in each vehicle, one being arranged for dining purposes and having accommodation for 12 passengers, the seats and tables being of the usual restaurant-car type, whilst the other portion of the car is arranged on similar lines to a buffet car and is fitted with fixed seats of a special design. These seats are upholstered in green leather and the seat adjacent to the gangway is hinged to facilitate access to the tables. There is accommodation for 18 seated passengers, whilst further standing room is available at a small bar.

The floor of the buffet portion is

Inside the restaurant section of the car.

Buffet compartment with bar in background.

covered with « Gesco » cork tiles and the screen separating the buffet portion from the dining saloon is provided with « Perspex » panels so that the staff may have a clear view the full length of the car.

The walls of the saloons are finished in coloured Rexine relieved by bands of anodised aluminium.

A large combined kitchen and pantry is provided for service to both saloons, the cooking being carried out entirely by electricity. The Stone-Wilson cooking stove provided in the kitchen comprises a roasting oven, three boiling plates and two grills. A separate fish fryer is provided, together with a Still's automatic boiler for the provision of tea and coffee, and a large electric refrigerator.

The power is derived from two 10-kilowatt axle-driven generators suspended beneath the vehicle, and an Exide-Ironclad battery of 210 ampere-hours capacity provides current whilst the train is standing at stations.

The vehicles are of the L. N. E. standard teak construction and are fitted with buckeye couplers, Pullman vestibules, and are mounted on L. N. E. standard compound bolster bogies. The weight of the vehicles is 41 tons 11 cwts 3 qrs.

MISCELLANEOUS INFORMATION.

[625. 154 (.44)]

1. — Turntable extension carrier to accommodate long-wheelbase wagons,

by Mr. MARJOLLET,

Ingénieur principal adjoint à la Division du Mouvement, French National Railways Company, Eastern Area

(Revue Générale des Chemins de fer.)

Traffic demands and the necessity for improving equipment by increasing the useful load of wagons in comparison with their tare have caused railway companies to increase the size of their wagon stock. In the same way the number and capacity of privately-owned wagons have been constantly increased on account of the more advantageous rates applied to them.

The use of these wagons could not, however, be extended to the whole of the existing plant, since the increase in wagon capacity necessitated increasing the wheelbase of the wagons to such a degree that they became too long to use the existing turntables. There are at present in service four-wheeled wagons with a wheelbase of 5 to 6 m. (16' 5" to 19' 8") and sometimes more, whilst most of the turntables have a diameter not exceeding 4.50 m. (14' 9") and can thus only accommodate wagons with a wheelbase of less than 4 m. (13' 1 1/2").

To obviate these difficulties, some turntables have been replaced by larger circular turntables, or by bridge turntables, at a high cost, but in most cases the available space or neighbouring obstacles have not allowed the installation of a cumbrous plant of this nature.

The device described below, whilst having a lower installation cost than that of a new circular or bridge turntable — in some cases as much as 50 % less — provided a simple solution of the problem of turning long-wheelbase wagons on the existing tables.

Figure 1 shews the device as it was applied for the first time to a turntable at Paris-La Villette.

A samll auxiliary carrier or truck, called « Chariot-satellite » and constructed after the fashion of a traverser, with articulated runners and carrying rails, can, with one pair of wagon wheels resting on it, run on a path concentric with the turntable, whilst the other pair of wheels rests at any point on the table according to the wagon wheelbase. Synchronous rotation of the turntable and of the extension truck is ensured solely by the adhesion of the wagon placed astride on the two devices.

As the extension carrier is required to work on a curve of very short radius (approx. 5 m.

Fig. 1.

16' 5") whilst supporting a load of 15 to 20 tons, i. e. about half the weight of a loaded wagon, the running and guiding parts have had to be designed to reduce friction to a minimum and to facilitate movement as much as possible.

The extension carrier is carried on only four wide round-rimmed rollers without tyres or flanges, mounted on roller bearings and set at an angle to work over the specified are of a circle on the flat iron runways concentric with the turntable.

These runways, which are set on a concrete base, are situated at the same level as the track rails so that the four rollers, by reason of their design and arrangement, can negotiate without mishap the gaps between the rail and the stock rail at the crossings.

The carrier is guided by an arrangement which constitutes one of the novelties of the system, i. e. is provided by two round steel rods which project vertically from the floor of the extension carrier on its centre line, and can rotate in the manner of a journal end in ball bearings fixed in the carrier frame (fig. 2).

These rods slide in a channel or guideway and have slight play. The guideway is concentric with the turntable and made up of two U irons, curved and stayed, and fixed in concrete, equidistant between the two runways; this channel is deep enough to remain effective when the carrier passes over the gap between rail and stock rail, and the guide rods are sufficiently long to remain constantly engaged and so exclude the possibility of a derailment.

The top of the guide channel is flush with the rail top, and the rails are slotted in one place only to allow the guide rods to pass.

In the case of a quarter-circle arrangement, that is to say for the accommodation of wagons on two roads at right-angles, which is usually limited by the exterior rails of the two roads served by the extension carrier, there are normally only two rails slotted, and at only one place. If it is desired to turn wagons right about, a semi-circular runway only need be provided.

In view of the fact that the carrier is not in any way an integral part of the turntable, it can be removed at will : thus there remains no obstacle on the line when the carrier is not in use, and the turntable may be used in the normal way for wagons of small wheelbase.

When not in use the carrier is generally left on the runway between the angle of the two roads if the location allows this to be done (fig. 1).

When the carrier can not be left here, however (for example, in the presence of other roads as shewn in the foreground in figure 3) the carrier can be stationed at any convenient point by running it on four slightly superelevated auxiliary rollers on two supplementary runways, also superelevated, which may, as in the case shown in figure 3, describe a curve in the opposite direction of that of the normal runway. This curve is provided by the channel guideway, which may be bent if required (see fig. 4), and the carrier remains guided over its whole course by the two guide rods.

The length of the carrier and the radii of its runways depend on the wheel-bases of the wagons to be turned. The installation at Paris-La Villette includes a carrier of 1 m. (3' 3 3/8") span, and the guideway is set at 5.25 m. (17' 2") from the pivot of the turn-

Half elevation. *Cross section.*

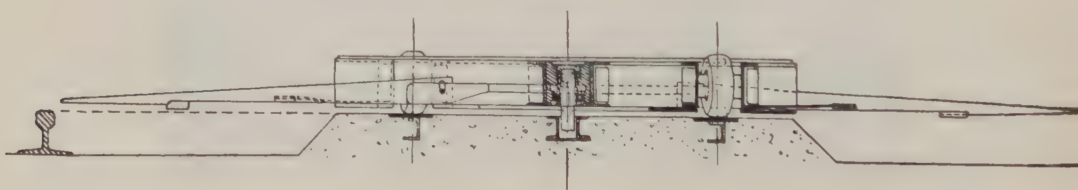


Fig. 2.

Fig. 3.

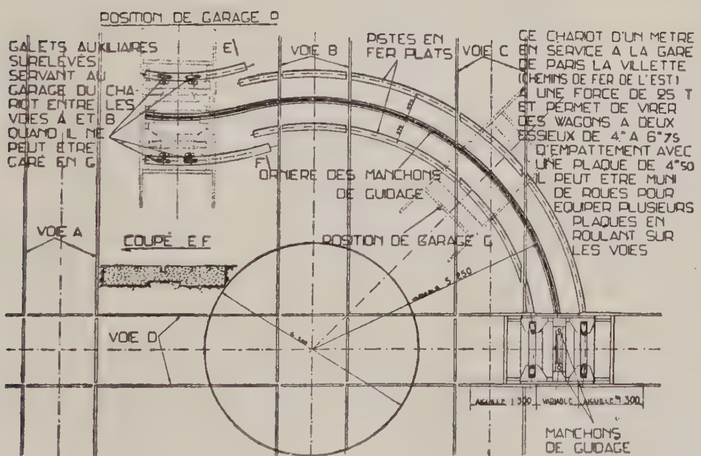


Fig. 4. — Extension carrier for turntable.

Explanation of French terms :

Position de garage = parking position. — Voie = track. — Pistes en fers plats = runway made of iron flats. — Ornière des manchons = channel for guiding rollers. — Galets auxiliaires... = auxiliary super-elevated rollers for parking the carrier between roads A and B, when it cannot be left at G. — Ce chariot d'un mètre... = this 1-m. (3' 3 3/8'') carrier in use at Paris-La Villette station carries a load of 25 tons and allows the turning of 4-wheeled wagons of 4 to 6.75 m. (22' 3'') wheelbase on a table of 4.5 m. (14' 9'') diameter. It may be provided with wheels to run on rails, and so be available for several tables.

table, which is of 4.50 m. (14' 9") diameter. The table can only turn wagons of 3.90 m. (12' 9 1/2") wheelbase or less, whilst the

Fig. 5.

combination of turntable and carrier will accommodate four-wheeled wagons of 3.90 m. to 6.75 m. (22' 3") inclusive.

The turning operation, using the carrier, does not involve the use of any more time or labour than with the table alone. The operation may be done manually by pushing the carrier end of the wagon and this is sufficient to assure the synchronous turning of the table and carrier (fig. 5).

Bogie wagons may also be turned with the same device. By means of a carrier of 3.50 m. (11' 6") span, with the guideway situated at 6.65 m. (21' 10") from the pivot of a 4.50 m. (14' 9") diameter table, all wagons, four or eight-wheeled, of up to 9.80 m. (32' 2") wheelbase could be turned. At the same time, to complete the arrangement for turning bogie wagons, it is necessary to provide a very simple device for rendering the wagon body rigid with the turntable, to prevent the body displacing itself obliquely in relation to the bogie which is resting on the table.

There are at present five extension carriers in service, two of which have been in use for three years at Paris-La Villette, where they are in daily use for turning many wagons. Their maintenance is cheap and simple, and their working has always given satisfaction.

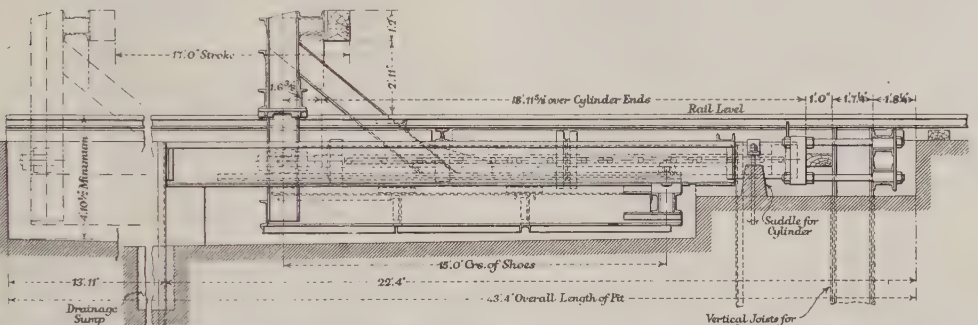
[625. 156 (.42)]

2. — Hydraulic buffer stop at Aldgate East Station.

(*Engineering.*)

In the issue of November 4, 1938, page 531, of *Engineering* a description was given of the reconstruction of the Aldgate East station of the London Passenger Transport Board. In connection with this work a new buffer stop, the first of a new type, is in course of manufacture at the works of Messrs. Ransomes and Rapier, Limited, Waterside Ironworks, Ipswich. The new buffer stop, which is illustrated in figs. 1 to 4, works on the well-established hydraulic principle in which the resistance is obtained by the movement of a piston in a cylinder containing fluid, oil being used in the present instance. The outstanding feature of the new stop is that an unusually long stroke is obtained. With the ordinary type of buffer stop, in which the piston rod is in compression during the retardation of the train, the stroke is limited to a maximum length of about 11 ft., and as a stroke of 17 ft. was required in the present instance, the installation was designed with the piston rod in tension. The buffers have a maximum duty of stopping a train weighing 300 tons and moving at 13 m.p.h. The resistance, which is practically constant throughout the stroke, is 106 tons. The arrangement of the installation is shown in figs. 1 to 3. It will be seen that the buffer stop is built into a pit 4 ft. 10 1/2 in. deep, and that it consists of a braced buffer structure, provided at one end with shoes which slide along the track rails, the vertical reaction being taken by a slipper on the underside of the pitside girders. These girders are 22 ft. 4 in. long, and they carry the track rails over the span of the open pit on their upper side. As shown in figs. 1 and 2, the buffer structure is con-

Fig. 1.



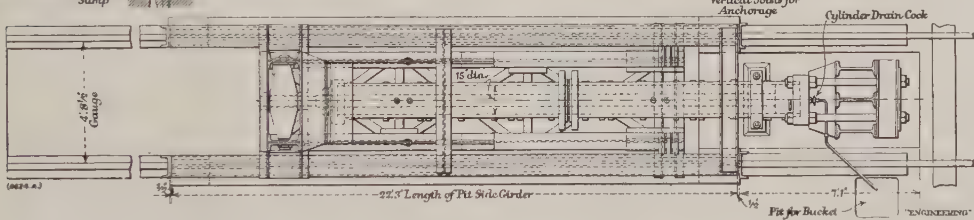


Fig. 2.

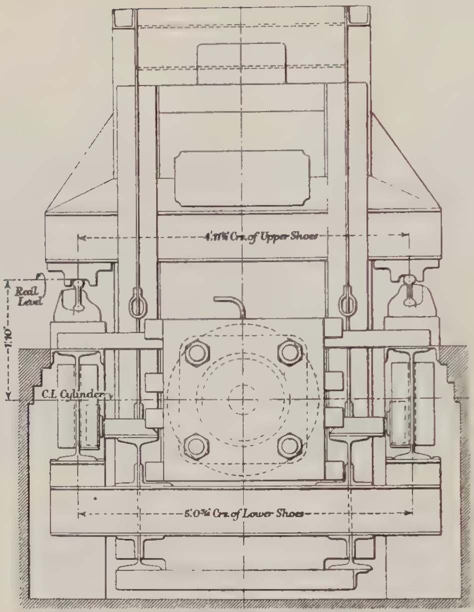


Fig. 3.

nected to a piston rod with the piston working in a hydraulic cylinder of sufficient length to accommodate the full stroke of 17 ft. The cylinder is securely anchored, as it has to resist the full retardation force of the buffer. The anchorage, shown in figs. 1 to 3, consists of four 3-in. diameter bolts with suitable cross girders connected to three vertical joists built into the foundation. Reinforcement below the pit level is provided for securing these girders. The cylinder is also supported by a saddle and strap shown in fig. 1, and by a cross girder between the two pitside girders towards the front end.

The cylinder is 15 in. in internal diameter, and consists of two lengths of solid drawn steel tube joined at the centre by means of flanges screwed on and bolted together. The cylinder tubes are also screwed at the rear end to receive the anchorage, and at the front end to receive a gland, fitted with a U-shaped leather, as shown in fig. 4, which prevents leakage along the piston rod. The rod itself is connected to the buffer structure by a swivel joint, shown in figs. 1 and 2, to relieve it of all bending stresses. The piston is provided with two longitudinal rectangular slots, which work over two tapered strips fixed inside the cylinder. Due to the taper of the strips, a diminishing area for the passage of the oil from the front to the back of the piston is provided throughout the stroke. The area of

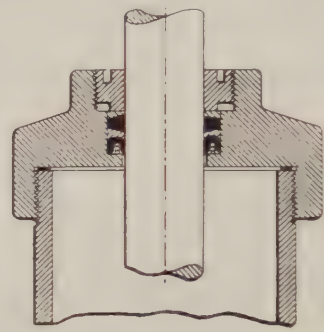


Fig. 4.
Section through cylinder end cover.

the orifices is so designed that the retardation force is constant. The cylinder is provided with a plug, and with air-release cocks, for use when filling or replenishing the light oil used. The centre line of the cylinder is 1 ft. 10 in. below rail level, and approximately 5 ft. 4 in. below the buffer level, so that considerable overturning effect will occur on impact by the train. For this reason, the buffer structure has a length of 15 ft. between the centre of the upper shoe, running on the track rail, and the centre of the lower shoe, running on the underside of the pitside girders. As shown in the figures, the structure is well braced and gusseted to resist distortion on impact, and a timber buffer is fitted on the cross girder. Automatic resetting of the stop was not required on this installation, and shackles are therefore fitted to the diagonal supports, at the rail level, as shown in figs. 1 and 2, for the attachment of rope tackle. The overall length of the pit for the installation is 43 ft. 4 in.

[385. (06 111]

OFFICIAL INFORMATION.

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(July 6th, 1939).

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(2) Retires at the 15th session.
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Kradolfer (3), directeur de l'Office fédéral des Transports de la Confédération suisse ; Berne ;

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(1) Retires at the 14th session.
(2) Retires at the 15th session.
(3) Retires at the 16th session.

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H. E. **Tarraf Aly** Bey [2], under secretary of State, Egyptian Ministry of Communications; Cairo;

R. E. **Thomas**, chief inspecting engineer (Egyptian Government), London;

Antonio **Valenciano y Mazerès** [1], inspecteur général des Ponts et Chaussées, administrateur de la Compagnie des Chemins de fer de Madrid à Saragosse et à Alicante; 5-3º, General Oraâ, Madrid (*);

Th. M. B. **van Marle** [2], inspecteur-generaal van het Verkeer, Rijksverkeersinspectie; The Hague;

L. **Velani** [3], directeur général des Chemins de fer de l'Etat italien; Villa Patrizi, Rome;

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(1) Retires at the 14th session.
(2) Retires at the 15th session.
(3) Retires at the 16th session.
(*) Situation at the 11th July, 1936.

A. M. M. **Verkoyen** (2), directeur du Service du Matériel de la Société Nationale des Chemins de fer belges; 17, rue de Louvain, Brussels;

Dr. C. C. **Wang** (1), representative of the Chinese Ministry of Railways; 21, Tothill street, Westminster, London, S. W. 1;

D. **Willard** (3), chairman of the Board, Association of American Railroads; president Baltimore & Ohio Railroad; Baltimore, Md.;

M. **Yoyitch** (1), directeur général adjoint des Chemins de fer de l'Etat yougoslave; Belgrade;

N... (3) (Brazil).

SECRETARY'S OFFICE : 231, rue Royale, Brussels.

General secretary : P. **Ghilain** (already named).

Assistant secretaries : A. W. **Chantrell**, ingénieur principal au Service du Matériel de la Société Nationale des Chemins de fer belges;

J. **Dubus**, ingénieur principal au Service de la Voie de la Société Nationale des Chemins de fer belges;

E. **Minsart**, ingénieur principal au Service du Matériel de la Société Nationale des Chemins de fer belges.

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